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A PRACTICAL TREATISE ON HEAT,

AS APPLIED TO THE USEFUL ARTS,

FOR THE USE OF

ENGINEERS, ARCHITECTS, ETC.

BY THOMAS BOX,

AUTHOR OF 'PRACTICAL HYDRAULICS,' 'MILL-GEARING,' ETC.

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PREFACE TO THE FIRST EDITION.

THE object of the following work is to give Data, Rules, and Tables to facilitate the practical application of the Laws of Heat to the Useful Arts.

The subject has throughout been largely illustrated by Examples worked out in detail, and this has led to calculations more or less complicated; but the author's special desire to make the "principles" of the subject clear to the reader could not be so well attained by any other means.

The authorities from whom the Experimental Data, &c., are derived, are for the most part given as they occur; but Péclet's great work, 'Traité de la Chaleur,' should be more particularly mentioned.

BATH, *August*, 1868.

PREFACE TO THE SECOND EDITION.

This new Edition is for the most part based on the former one, but considerable additions have been made throughout, especially in the chapters on Evaporation, Heating Liquids and Air, Ventilation, &c., and some errors have been corrected. A copious Index has also been added, which, it is hoped, will render the work more convenient as a book of reference, and more generally useful.

Sandown, *August*, 1876.

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PRACTICAL TREATISE ON HEAT.

CHAPTER I.

GENERAL PRINCIPLES AND DATA.

(1.) "*Unit of Heat.*"—It is necessary to have a standard for measuring the amount of heat absorbed or evolved during any operation: in this country the standard "unit" is the amount of heat required to raise the temperature of a pound of water at 32°, one degree Fahrenheit. It is necessary thus to restrict it to 32°, because the specific heat of water varies slightly with the temperature, as shown by (3).

(2.) "*Specific Heat.*"—Different bodies require very different quantities of heat to effect in them the same change of temperature. The capacity of a body for heat is termed its "Specific Heat," and may be defined as the number of units of heat necessary to raise the temperature of 1 lb. of that body 1° Fahr. Table 1 gives its value for most of the solid and liquid bodies commonly met with in practice, and shows that it is very variable; for instance, mercury requires only $\frac{1}{30}$th of the heat necessary for the same weight of water. The specific heat of all bodies (except gases) increases with the temperature; this is shown by Table 2, from the experiments of Dulong and Pouillet. With a variable rate it is necessary to distinguish the specific heat *between* two given temperatures from that *at* a given temperature, as explained for variable expansion in (26). At ordinary temperatures the departure from uniformity is not great; for instance, Table 2 shows that at the high temperature of 2192° the specific heat of platina is only 14 per cent. greater than at 212°, and in most cases we may admit a uniform rate having the value given by Table 1. Thus, to heat 200 lbs. of cast iron

120° would require 200 × 120 × ·12983 = 3116 units of heat; the same weight of water requiring 200 × 120 = 24000 units, or nearly eight times the heat required for cast iron.

TABLE 1.—Of the SPECIFIC HEAT of SOLID and LIQUID BODIES, being the number of Units of Heat necessary to heat One Pound of the Body 1° Fahrenheit.

	Specific Heat.	Authority.
METALS—		
Cast Iron	·12983	Regnault.
Wrought Iron	·11379	"
Zinc	·09555	"
Copper	·09515	"
Brass	·09391	"
Silver	·05701	"
Tin	·05695	"
Mercury	·03332	"
Gold	·03244	"
Platina	·03243	"
Lead	·03140	"
Bismuth	·03084	"
WOODS—		
Pine	·650	Mayer.
Oak	·570	"
Birch	·480	"
EARTHS, &c.—		
Ice	·504	Person.
Beeswax	·450	Gadolin.
Spermaceti	·320	Irvine.
Coal	·2777	Crawford.
Marble	·21585	Regnault.
Chalk	·21485	"
Sulphur	·20259	"
Coke	·20085	"
Glass	·19768	"
Burnt Clay	·185	Gadolin.
LIQUIDS—		
Water at 32°	1·000	..
Alcohol (sp. gr. ·793) ..	·622	Dalton.
Ether, Sulphuric	·477	Regnault.
Oil of Turpentine	·472	Despretz.
Petroleum	·434	Regnault.
Olive Oil	·3096	Lavoisier.

(3.) The specific heat of water varies slightly with the temperature, as shown by the experiments of Regnault in

TABLE 2.—Of the VARIATION in the SPECIFIC HEAT of BODIES at different Temperatures.

	Between 32° and 212°.	Between 32° and 392°.	Between 32° and 572°.
Mercury (Dulong)	·0330	..	·0350
Silver "	·0557	..	·0611
Zinc "	·0927	..	·1015
Copper "	·0940	·1013	..
Iron "	·1098	·1150	·1218
Platina "	·0335	..	·0355
Antimony "	·0507	..	·0547
Glass "	·1770	..	·1900

	At 212°.	At 572°.	At 932°.	At 1292°.	At 1832°.	At 2192°.
Platina (Pouillet) ..	·03350	·03434	·03518	·03602	·03728	·03818

Table 3. Thus, at 446° it is about $5\frac{5}{8}$ per cent. greater than at 32°; the difference is small, and for most practical purposes may be neglected, but where great accuracy is required for the purposes of science, it should be remembered and allowed for.

Liquids differ less in specific heat among themselves than solids, being seldom less than one-third that of water: in all probability the specific heat of all liquids increases with the temperature, as is the case with water, but of that we have no experimental evidence.

Dulong has shown that the specific heat and the atomic weight of simple bodies are inversely proportional to each other, so that their product is in all cases constant, and this law has been confirmed by Regnault. In Table 4 the specific heat is taken from Table 1 and the atomic weights from Table 40, and the third column shows that their product is constant, or nearly so, the mean of the whole being 40. This law is useful by enabling us to find the specific heat from the atomic weight, or *vice versâ.* Thus we do not know the specific heat of aluminium, but by Table 40 its atomic weight is 171·16; therefore by the law of Dulong its specific heat will be 40 ÷ 171·16 = ·2337: similarly for nickel we obtain 40 ÷ 369·67 = ·1082; Regnault's experiments give ·10863.

TABLE 3.—Of the SPECIFIC HEAT of WATER at different Temperatures, according to the Experiments of REGNAULT.

Temperature by an Air-Thermometer. Fahr.	Difference from 32°.	Units of Heat to raise 1 lb. of Water from 32° to the given Temperature.	Mean Specific Heat between 32° and the given Temperature.	Specific Heat at the given Temperature.
32°	0	0·0000	..	1·0000
50	18	18·0036	1·0002	1·0005
68	36	36·0180	1·0005	1·0012
86	54	54·0468	1·0009	1·0020
104	72	72·0918	1·0013	1·0030
122	90	90·1546	1·0017	1·0042
140	108	108·2466	1·0023	1·0056
158	126	126·3780	1·0030	1·0072
176	144	144·5076	1·0035	1·0089
194	162	162·6858	1·0042	1·0109
212	180	180·9000	1·0050	1·0130
230	198	199·1538	1·0058	1·0153
248	216	217·4508	1·0067	1·0177
266	234	235·7946	1·0076	1·0204
284	252	254·1870	1·0087	1·0232
302	270	272·6316	1·0097	1·0262
320	288	291·1338	1·0109	1·0294
338	306	309·6936	1·0121	1·0328
356	324	328·3164	1·0133	1·0364
374	342	347·0022	1·0146	1·0401
392	360	365·7600	1·0160	1·0440
410	378	384·5880	1·0174	1·0481
428	396	403·4916	1·0189	1·0524
446	414	422·4744	1·0204	1·0568
(1)	(2)	(3)	(4)	(5)

(4.) "*Specific Heat of Air and Gases.*"—When air is heated in a closed vessel, the volume remaining constant, the pressure is increased; if, on the contrary, the air is suffered to expand freely by increase of temperature, the pressure may remain constant. Now, the specific heat of air, or the amount of heat required to effect a given change of temperature, is different under these two conditions, for reasons that will presently appear.

As the basis of exact calculation, we may admit from Regnault's experiments, that a litre of air at 32° Fahr., under a pressure of ·76 metre of mercury in the barometer also at 32°, weighs 1·293187 gramme, and that a litre of mercury at

TABLE 4.—Of the ATOMIC WEIGHTS of METALS multiplied by their SPECIFIC HEAT.

Metal.	Specific Heat, S. Regnault.	Atomic Weights, W.	Product, S × W.
Wrought Iron	·11379	339·2	38·60
Zinc	·09555	403·23	38·53
Copper	·09515	395·69	37·65
Tin	·05695	735·29	41·60
Gold	·03244	1243·01	40·32
Platina	·03243	1233·5	40·00
Lead	·03140	1294·5	40·65
Bismuth	·03084	1330·37	41·03
Sulphur	·20259	201·16	40·75
Mean	..	..	40·00

32° weighs 13595·93 grammes. Reducing these to English measures, the metre being 39·3708 inches, we have 39·3708 × ·76 = 29·922 inches of mercury in the barometer. Then, a gramme being ·00220462 lb., and a litre 61·02711 cubic inches, the weight of a cubic inch of mercury will be 13595·93 × ·00220462 ÷ 61·02711 = ·49116 lb., and a column 29·922 inches high will exert a pressure of ·49116 × 29·922 = 14·696 lbs. per square inch, or 14·696 × 144 = 2116·2 lbs. per square foot. A litre being ·035317 cubic foot, a cubic foot of air at 32° weighs 1·293187 × ·0020462 ÷ ·035317 = ·080726 lb., and 1 lb. of air measures 1 ÷ ·080726 = 12·387 cubic feet.

(5.) "*Specific Heat with Constant Pressure.*"—Imagine a gas-holder so well balanced as to exert no pressure on the enclosed gas beyond that due to the pressure of the atmosphere, and let it contain 1 lb. of air, or 12·387 cubic feet at 32°. If heat be applied so as to raise the temperature 1° or to 33°, the air would be expanded slightly, and the gas-holder would rise to allow of that expansion. The amount of heat required under these circumstances, according to Regnault's experiments, would be ·2379 unit, or the amount that would have raised the temperature of the same weight of water ·2379 of a degree. Table 5 gives in col. 1 the amount for other gases under similar conditions.

TABLE 5.—Of the SPECIFIC HEAT of GASES and VAPOUR.

Relative Specific Gravity. Air = 1·0.	Gas, &c.	Apparent Specific Heat with Pressure Constant by Regnault's Experiments.	Normal Specific Heat with Volume Constant calculated.	Ratio of Specific Heat under Constant Pressure to Specific Heat with Volume Constant.	
				Calculated.	By Dulong's Experiments.
1·00000	Atmospheric Air ..	·2379	·16866	1·4105	1·421
1·10563	Gas, Oxygen	·2182	·15558	1·402	1·415
0·06926	„ Hydrogen ..	3·4046	2·4046	1·416	1·407
0·97137	„ Nitrogen ..	·2440	·17272	1·412	..
1·52901	„ Carbonic Acid	·2164	·17112	1·264	1·338
0·96740	„ Carbonic Oxide	·2479	·17633	1·406	1·427
1·17488	„ Sulphuretted Hydrogen ..	·2423	·18336	1·321	..
2·45307	„ Chlorine	·1214	·09317	1·303	..
0·62350	Vapour of Water ..	·4750	·36400	1·305	..
		(1)	(2)	(3)	(4)

(6.) "*Specific Heat with Constant Volume.*"—In the case we have just considered, the ·2379 unit of heat not only raised the temperature of the pound of air 1°, but also did a certain amount of *mechanical work* in raising the gas-holder against the pressure of the atmosphere, and this was unavoidable, because the essential condition assumed, namely, that the pressure should be constant, necessitated increase of volume by expansion and the expenditure of a certain portion of heat in performing mechanical work.

Say that our gas-holder, holding 1 lb. of air at 32°, was 1 foot square on plan, therefore 12·387 feet high, then the pressure of the atmosphere on the top of it would be 2116·2 lbs., as we have seen in (4), and that weight has to be raised a certain distance when the gas-holder rises by the air expanding. By the rule in (27) the volume at 33° would become $12 \cdot 387 \times \frac{458 \cdot 4 + 33}{458 \cdot 4 + 32} = 12 \cdot 41226$ cubic feet, so that the gas-holder rises with its load $12 \cdot 41226 - 12 \cdot 387 = \cdot 02526$ foot, and we have $\cdot 02526 \times 2116 \cdot 2 = 53 \cdot 455$ foot-pounds of work done. By Joule's experiments (46) the mechanical equivalent of a unit of heat is 772 foot-pounds, hence the work done in our

case requires $53 \cdot 455 \div 772 = \cdot 06924$ unit of heat. But if the air had been heated in a closed vessel this mechanical work would not have been done at all, expansion being prevented, and the pound of air would then have required only $\cdot 2379 - \cdot 06924 = \cdot 16866$ unit of heat to raise its temperature 1°, hence $\cdot 16836$ is the specific heat of air with constant volume, that with constant pressure being $\cdot 2379$. The ratio is therefore $\cdot 2379 \div \cdot 16866 = 1 \cdot 4105$ to $1 \cdot 0$.

(7.) Again, with hydrogen the specific heat with constant pressure is $3 \cdot 4046$ by Table 5, and the volume of 1 lb. at 32° by col. 1 of Table 39 is $12 \cdot 387 \div \cdot 06926 = 178 \cdot 87$ cubic feet. At 33° its volume would become

$$178 \cdot 87 \times \frac{458 \cdot 4 + 33}{458 \cdot 4 + 32} = 179 \cdot 23474 \text{ cubic feet,}$$

and the gas-holder 1 foot square on plan and $178 \cdot 87$ feet high, rises $179 \cdot 23474 - 178 \cdot 87 = \cdot 36474$ foot; and the atmospheric pressure being as before $2116 \cdot 2$ lbs., the mechanical work done by the heat is $\cdot 36474 \times 2116 \cdot 2 = 771 \cdot 86$ foot-pounds, which is equal to $771 \cdot 86 \div 772 = 1$ unit of heat nearly. With volume constant we should only require $3 \cdot 4046 - 1 \cdot 0 = 2 \cdot 4046$ units of heat to raise the temperature of a pound of hydrogen 1°; the ratio is $3 \cdot 4046 \div 2 \cdot 4046 = 1 \cdot 416$ to 1.

(8.) From this we get a rule applicable to all gases by which the specific heat with constant volume may be found from the known specific heat with constant pressure, and the specific gravity of the gas, then

$$S' = S - \frac{\cdot 06924}{G},$$

in which S = the specific heat with constant pressure, S' with constant volume, and G = the specific gravity of the gas, air being $1 \cdot 0$. Thus, with hydrogen, in which $S = 3 \cdot 4046$ by Table 5, and $G = \cdot 06926$ by Table 39, we have

$$S' = 3 \cdot 4046 - \frac{\cdot 06924}{\cdot 06926} = 2 \cdot 4046$$

nearly as before. By this rule we have obtained cols. 2 and 3 in Table 5, taking as a basis the experimental specific heat under

constant pressure in col. 1. In col. 4 we have given the ratios found by the experiments of Dulong.

(9.) "*Specific Heat with variable Pressure and Volume.*"—It is important to observe that essentially the *normal* specific heat of air is that with volume constant or ·16866. By extraneous circumstances of pressure, &c., the apparent specific heat may be much greater than that, but may always be found by adding to the normal specific heat the extra amount due to the mechanical work done.

Thus, say we take as before a pound or 12·387 cubic feet of air, and find the amount of heat necessary to double its volume not under the constant pressure of 2116·2 lbs. per square foot, but under a progressively increasing one. Say that instead of a dead weight we had a huge spiral spring giving 2116·2 lbs. with the normal volume at 32°, and double pressure or 4232·4 lbs. when the volume was doubled by the gas-holder rising 12·387 feet. In this case, therefore, neither the pressure nor volume would be constant. To double both the pressure and volume of air at 32°, its temperature must be increased 1471°·2, or from 32° to 1503°·2; by the rule in (27) the volume would then become $12 \cdot 387 \times \frac{2116 \cdot 2}{4232 \cdot 4} \times \frac{458 \cdot 4 + 1471 \cdot 2}{458 \cdot 4 + 32} = 24 \cdot 774$ cubic feet. Now, on the principle just stated, merely to heat the pound of air 1471°·2 we should require $1471 \cdot 2 \times \cdot 16866 =$ 258·1 units of heat; to this has to be added the heat required to do the mechanical work. The pressure increasing throughout the stroke or lift of the gas-holder in arithmetical progression, its mean is $(2116 \cdot 2 + 4232 \cdot 4) \div 2 = 3174 \cdot 3$ lbs., and the gas-holder rising with that mean load 12·387 feet, we have $12 \cdot 387 \times 3174 \cdot 3 = 39320 \cdot 2$ foot-pounds, requiring $39320 \cdot 2 \div 772 = 50 \cdot 93$ units of heat; the total is $258 \cdot 1 + 50 \cdot 93 = 309 \cdot 03$ units. In that case, as we require 309·03 units to heat a pound of air 1471°·2, the apparent specific heat is $309 \cdot 03 \div 1471 \cdot 2 = \cdot 210$.

(10.) Again, say that we have an arrangement such that as the gas-holder rose the pressure upon it was *diminished* to half the normal amount when the volume was increased from 1 to 2·1, or to $12 \cdot 387 \times 2 \cdot 1 = 26 \cdot 0127$ cubic feet. We should

have to increase the temperature only $24°\cdot52$, or from $32°$ to $56°\cdot52$; by the rule in (27) the volume would then become $12\cdot387 \times \frac{2116\cdot2}{1058\cdot1} \times \frac{458\cdot4 + 56\cdot52}{458\cdot4 + 32} = 26\cdot0127$ cubic feet, and the gas-holder would rise $26\cdot0127 - 12\cdot387 = 13\cdot6257$ feet. To heat the air only we require $24\cdot52 \times \cdot16866 = 4\cdot13$ units; the mean pressure is $(2116\cdot2 + 1058\cdot1) \div 2 = 1587\cdot15$ lbs., and the mechanical work done by the heat $13\cdot6257 \times 1587\cdot15 = 21726$ foot-pounds, which is equal to $21726 \div 772 = 28\cdot13$ units of heat; the total is $28\cdot13 + 4\cdot13 = 32\cdot26$ units. Hence the apparent specific heat in this case is $32\cdot26 \div 24\cdot52 = 1\cdot311$, which is $1\cdot311 \div \cdot16866 = 7\cdot77$ times the normal specific heat with volume constant.

(11.) The ratio of the specific heat with constant pressure to that with constant volume may also be determined from the velocity of sound. It was shown by Newton that the velocity with which vibrations are propagated by elastic fluids is the same as that of a body falling freely by gravity in a vacuum through *half* the height of a homogeneous atmosphere (148), having throughout the same density as at the surface of the earth. By the data given in (4) the density of air at $32°$ is $1\cdot293187 \div 13595\cdot93 = \cdot0000951$, that of mercury being $1\cdot0$, and the height of the barometer being $29\cdot922$ inches or $2\cdot4935$ feet, we obtain $2\cdot4935 \div \cdot0000951 = 26220$ feet as the height of the homogeneous atmosphere, the half of which is 13,110 feet, and the velocity due to that height by the laws of falling bodies is $\sqrt{13110} \times 8\cdot025 = 918\cdot8$ feet per second. But by Table 6 the mean of six of the best observations on the velocity of sound gives $1089\cdot2$ feet per second, the height due to which is $(1089\cdot2 \div 8\cdot025)^2 = 18420$ feet instead of 13,110 feet as due to theory. Laplace has suggested that this difference is due to the heat generated by the compression (47) which causes the vibration, and that in effect the half-height of the homogeneous atmosphere is increased in the ratio of the specific heat with constant pressure to that with constant volume: admitting this, we find that ratio to be as 1 to $18420 \div 13110 = 1\cdot405$, which is very nearly $1\cdot4105$ as calculated in (6).

TABLE 6.—Of the RESULTS of EXPERIMENTS on the VELOCITY of SOUND in Air at 32°.

Observers, &c.	Distance between Stations.		Observed Velocity in Feet per Second.	Height in Feet due to observed Velocity, H.	Half-height of Homogeneous Atmosphere, h.	Ratio, $\frac{H}{h}$.
	Feet.	Miles.				
Benzenberg ..	29764	5·637	1093·0	18982	13110	1·415
Goldingham	29547	5 596	1086·7	18337	13110	1·398
Myrbach ..	32615	6·177	1092·1	18520	13110	1·413
Arago, Prony, Humboldt	61064	11·565	1086·1	18317	13110	1·397
Gregory ..	13460	2·549	1088·05	18380	13110	1·402
Van-Beek, &c.	57839	10·954	1089·42	18429	13110	1·406
Mean of the whole ..	..	..	1089·23	18420	13110	1·405

(12.) "*Liquefaction.*"—When metals, ice, &c., are heated to a certain temperature they "melt," or pass from the solid to the liquid state; when the process is reversed and liquids are cooled down to a certain point they "freeze," or pass from the liquid to the solid state. The melting and freezing points are therefore identical; thus the melting point of ice and the freezing point of water are both = 32°. Table 7 gives the melting points of metals, &c., from the experiments of Pouillet; the high temperatures were measured by an air-thermometer, and the results differ very much from those given in the old tables of Wedgwood, Morveau, and Daniell, which were obtained by other and less correct means (34).

(13.) "*Latent Heat of Liquefaction.*"—When a body passes from the solid to the liquid state it absorbs a large amount of heat without changing its own temperature, the heat thus absorbed becoming latent and insensible to the thermometer. This is termed the "latent heat of liquefaction," and may be defined as the number of units of heat absorbed by 1 lb. of the solid in passing to the liquid state. When the process is reversed, the liquid congealing, freezing, or passing back again to the solid state, the same amount of heat is emitted or restored. Thus, when ice is heated to 32° it begins to melt, the temperature remaining

Table 7.—Of the Melting Points of Metals, &c., according to the Experiments of Pouillet; the temperatures above dull red heat were measured by an Air-Thermometer.

	Fahr.	
Wrought Iron, English, hammered ..	2910°	M. Pouillet.
„ French, soft	2730	„
Steel, maximum	2550	„
„ minimum	2370	„
Cast Iron, grey, 2nd fusion	2190	„
„ „ very fusible	2010	„
„ white, maximum	2010	„
„ „ minimum	1920	„
Gold, very pure	2280	„
„ standard coin	2156	„
Copper	2050	„
Silver, very pure	1830	„
Brass	1650	„
Antimony	810	„
Zinc	793	„
Lead	630	„
Bismuth	518	„
Tin	455	„
Sulphur	239	„
Wax, white	154	„
„ unbleached	143	„
Spermaceti	120	„
Stearine .. .. 109° to	120	„
Phosphorus	109	„
Tallow	92	„
Oil of Turpentine	14	„
Mercury	− 40	„
Bismuth, 4; Tin, 1; Lead, 1	201	„
„ 8 „ 3 „ 5	212	„
„ 5 „ 3 „ 2	212	„
„ 5 „ 4 „ 1	246	„
„ 1 „ 1 „ 0	286	„
„ 1 „ 2 „ 0	334	„
„ 0 „ 3 „ 1	367	„
„ 0 „ 4 „ 1	372	„
„ 0 „ 5 „ 1	381	„
„ 0 „ 2 „ 1	385	„
„ 1 „ 3 „ 0	392	„
„ 0 „ 1 „ 1	466	„
„ 0 „ 1 „ 3	504	„
Common Salt, 1; Water, 3 freezes	4	Dr. Ure.
Sulphuric Acid, sp. gr. 1·6415 „	− 45	„
„ Ether .. .. .. „	− 46	„
Mercury (by Air-Thermometer) „	− 37·9	B. Stewart.

fixed until the whole of the ice is melted, and during this process 142·4 units of heat are absorbed, being the amount that would have raised the same weight of water 142°·4; but the ice itself having by Table 1 a specific heat of ·504 would have had its temperature raised $142 \cdot 4 \div \cdot 504 = 281°$ or to 313° if the heat had not become latent. The latent heat of liquefaction, however, is not 281 but 142·4 *units*. Table 8 gives the latent heat for many bodies, the third column being obtained by dividing the latent heat by the specific heat in Table 1.

This property of *fixedness* in the melting point of solids is a valuable one; the melting point of ice is used for obtaining one of the standard points in graduating thermometers (38), but some care is necessary to obtain correct results, the ice should be pounded fine or, better still, snow should be used, and the bulb must be completely enveloped in it so as not to touch the bottom or sides of the vessel; without these precautions an error of one or two degrees may occur.

In tempering steel instruments (37), where, in order to secure uniformity of temper in the whole of an article of considerable size, uniformity of heat is both essential and difficult to obtain, metallic baths are commonly used, consisting of an alloy of lead and tin, &c., which melts at the particular temperature desired. By keeping always a small portion unmelted, the whole mass may be maintained at the temperature due to liquefaction without the trouble and uncertainty of regulating the fire. Table 9 gives the melting points of alloys for a great range of temperatures; others are given by Table 7.

(14.) "*Ebullition.*"—When the temperature of any liquid is raised to a certain point it "boils" or passes off in a state of vapour, and the temperature of the liquid remains constant until the whole is vapourized. This temperature varies with the character of the liquid, and also with the pressure of the atmosphere, or (in the case of a closed vessel) with the pressure of the vapour on the surface of the liquid. Table 10 gives the boiling point of many liquids in open vessels with the ordinary atmospheric pressure of 30 inches of mercury in the barometer, and Table 11 gives in round numbers the

TABLE 8.—Of the LATENT HEAT of LIQUEFACTION, being the number of Units of Heat absorbed by One Pound of different Bodies in changing their state from Solid to Liquid.

	Latent Heat in Units.	Increase of Temperature in the Body if Heat had not been absorbed in melting.	Authority.
		°	
Ice to Water ..	142·4	281	Person.
Sulphur	16·8	83	"
Tin	25·6	450	"
Lead	9·7	309	"
Zinc	50·6	530	"
Bismuth	22·8	740	"
Silver	38·0	665	"
Cast Iron ..	233·0	1574	Clement.
Beeswax	78·7	175	Irvine.
Spermaceti ..	46·4	145	"

TABLE 9.—Of the MELTING POINTS of ALLOYS of BISMUTH, LEAD, and TIN.

Temp.	Bismuth.	Lead.	Tin.	Temp.	Lead.	Tin.
°				°		
202	8	5	3	380	4	22
208	8	6	3	390	5	4
220	8	7	3	400	11	8
230	8	8	3½	410	25	16
240	8	8	5	420	7	4
250	8	8	7	430	15	8
260	8	9	8	440	8	4
270	8	12	8	450	17	8
280	8	13	8	460	9	4
290	8	14	14	470	10	4
300	8	16	8	480	23	8
310	8	20	24	490	14	4
320	8	26	24	500	33	8
330	8	28	24	510	19	4
340	0	4	8	520	25	4
350	0	4	10½	530	30	4
360	0	4	13	540	38	4
370	0	4	17	550	48	4

variation under different pressures calculated from Regnault's experiments.

Mr. Dalton discovered the remarkable law that the *difference* of the temperatures of the boiling points of liquids is constant under all variations of pressure. Thus, under the ordinary atmospheric pressure, water boils at 212° and ether at 100°, as in Table 10; the difference is 212 − 100 = 112°. Now by Table 11 water under 6 atmospheres in col. 1 boils at 319° by col. 5, hence by the law of Dalton, ether should boil under that same pressure at 319 − 112 = 207°, which is within 1° of the temperature given by Regnault's experiment in col. 8. The col. 9 has been thus calculated throughout by subtracting 112° from the temperature in col. 5. With oil of turpentine the difference is 316 − 212 = 104° *greater* than water, it has therefore to be *added* to the temperatures in col. 5, and we have thus obtained the temperatures in col. 7 for pressures above the atmosphere where we had no experimental information. The experiments of Regnault have shown that this law of Dalton is only approximately true (184).

Tables 67 and 71 give also the temperatures of the boiling points of water at different pressures, the latter coinciding with the elastic force of vapour (183) at the same temperature.

TABLE 10.—Of the BOILING POINTS of LIQUIDS, at Atmospheric Pressure.

			Temp. Fahr.	
			°	
Ether, Sulphuric, sp. gr.		·7365 ..	100	G. Lussac.
Alcohol	"	·813 ..	173	Ure.
Muriatic Acid	"	1·047 ..	222	Dalton.
Nitric Acid	"	1·16 ..	220	"
Sulphuric Acid	"	1·3 ..	240	"
" "	"	1·85 ..	620	"
Oil of Turpentine	..	.. .. ..	316	Ure.
Naphtha	..	.. .. ..	306	"
Sulphur	..	.. .. ..	570	"
Linseed Oil ..	..	.. .. ..	600	..
Mercury	..	.. .. ..	662	Regnault.
Water	..	.. .. ..	212	..

TABLE 11.—Of the TEMPERATURE of the BOILING POINT of various LIQUIDS under different PRESSURES, deduced from the Experiments of REGNAULT.

Pressure.				Water.	Alcohol.	Oil of Turpentine.	Ether.	
Total, or above Vacuum.		Above the Atmosphere.					Experiment.	Calculated.
Atmosphere.	Inches of Mercury.	Inches of Mercury.	Lbs. per Sq. Inch.	Temperature of Ebullition.				
				°	°	°	°	°
$\frac{1}{6}$	5	—25	—12·2	134	101	210	20	22
$\frac{1}{3}$	10	—20	— 9·8	162	127	248	45	50
$\frac{1}{2}$	15	—15	— 7·3	180	144	271	62	68
1	30	0	0·0	212	173	316	100	100
2	60	30	14·7	249	206	353	132	137
3	90	60	29·4	273	228	377	157	161
4	120	90	44·0	291	245	395	177	179
5	150	120	58·8	306	258	410	192	194
6	180	150	73·5	319	270	423	206	207
7	210	180	88·2	330	280	434	217	218
8	240	210	103	339	289	443	228	227
9	270	240	117	348	297	452	237	236
10	300	270	132	357	305	461	246	245
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)

(15.) "*Boiling Point of Saline Solutions.*"—The boiling point is not affected by foreign bodies in the liquid so long as the body does not chemically combine with it; thus stones, masses of metal, &c., would not have any effect on the boiling point. But a great number of salts do combine chemically, and the boiling point is thereby raised as shown by Table 12, which gives the boiling points of *saturated* solutions; of course by dilution any temperature intermediate between 212° and that given in the table may be obtained. By the law of Dalton (14) the boiling point of any of the solutions under any other pressure may be easily found; thus under a *total* pressure of 6 atmospheres (29) chloride of calcium would boil at $319 + (355 - 212) = 462°$.

(16.) It should be observed that the *vapour* formed at the surface of a saline solution is that of pure water, and has the temperature of 212° at atmospheric pressures, although the temperature of the solution itself may be much higher. Thus a

TABLE 12.—Of the BOILING POINTS of SOLUTIONS of various SALTS, at ordinary Atmospheric Pressure, from the Experiments of M. LEGRAND and others.

	Temperature of Ebullition.	Weight of Salt in 100 lbs. of Water.
	°	lbs.
Saturated solution of Chlorate of Potassa	219·56	61·50
" " Carbonate of Soda	220·28	48·50
" " Phosphate of Soda	221·9	113·30
" " Chloride of Potassium ..	226·94	59·40
" " Common Salt	227·12	41·20
" " Neutral Tartrate of Potassa	238·40	296·20
" " Nitre, or Saltpetre	240·62	335·10
" " Nitrate of Soda	249·80	224·80
" " Acetate of Soda	255·90	209·00
" " Carbonate of Potash	275·00	205·00
" " Nitrate of Lime	303·80	362·20
" " Acetate of Potassa	336·20	798·20
" " Chloride of Calcium	355·10	325·00
" " Nitrate of Ammonia	356·00	Infinite

saturated solution of common salt boils at 227°, but the *steam* would have a temperature of 212° only. The fixedness of the temperature of steam has led to its adoption in determining one of the standard points in graduating thermometers. The Royal Commissioners for weights and measures and the British Association have agreed that the boiling point 212° shall be the temperature of steam at the pressure of 29·905 inches of mercury at 32°. Table 13, calculated from the experiments of Regnault, gives to one-tenth of a degree the corrections for small variations from the standard pressure.

(17.) "*Latent Heat of Vapourization.*"—By (14) it is shown that when water is raised to the boiling point, the temperature remains fixed, although the liquid is continuously receiving heat as before. The heat thus received is not sensible to the thermometer, for both the water and the steam have the constant temperature of 212°. The heat that thus disappears is termed the "latent heat of vapourization," to distinguish it from that of liquefaction (13), and it may be defined as the number of units absorbed by 1 lb. of a liquid in the act of passing into

TABLE 13.—Of the VARIATION in the BOILING POINT of WATER, with Variations in the Barometric Pressure, calculated from REGNAULT'S Experiments.

Temp. of Steam.	Pressure in Inches of Mercury at 32°.	Temp. of Steam.	Pressure in Inches of Mercury at 32°.	Temp. of Steam.	Pressure in Inches of Mercury at 32°.
°		°		°	
210·0	28·746	211·4	29·553	212·8	30·381
210·1	28·804	211·5	29·611	212·9	30·441
210·2	28·862	211·6	29·669	213·0	30·502
210·3	28·919	211·7	29·728	213·1	30·563
210·4	28·976	211·8	29·787	213·2	30·624
210·5	29·033	211·9	29·846	213·3	30·685
210·6	29·090	212·0	29·905	213·4	30·746
210·7	29·147	212·1	29·964	213·5	30·807
210·8	29·205	212·2	30·023	213·6	30·868
210·9	29·263	212·3	30·082	213·7	30·929
211·0	29·321	212·4	30·141	213·8	30·990
211·1	29·379	212·5	30·201	213·9	31·051
211·2	29·437	212·6	30·261	214·0	31·112
211·3	29·495	212·7	30·321	214·1	31·173

vapour. Thus a pound of water at 212° passing to steam at 212° absorbs, according to Regnault, as much heat as would have raised the temperature of the water 966° if it had not become latent. We have omitted here the consideration of the variable specific heat of water (3) in order to simplify the matter. Again, a pound of alcohol absorbs 457 units in the act of vapourization, or the amount of heat that would have raised a pound of water 457°; but alcohol having, by Table 1, a specific heat of ·622, its own temperature would have been raised 457 ÷ ·622 = 735° if heat had not become latent. Table 14 gives the latent heat of vapourization for a few of the liquids most commonly met with.

(18.) "*Effects of Pressure.*"—The temperature of the boiling point varies with the pressure, as shown by (14) and Table 11, and it has been found by experiment that the amount of heat which becomes latent during vapourization varies with the temperature at which it is effected, but that the total amount of heat necessary to raise the liquid from a low temperature and then evaporate it is constant, or nearly so. Thus, the heat required to raise a pound of water from 0° to 212° and then

TABLE 14.—Of the LATENT HEAT of VAPOURIZATION, being the number of Units of Heat required to convert Liquids from their respective Boiling Points to Vapour, under a Pressure of 30 inches of Mercury.

	Latent Heat in Units.	Increase of Temperature of Liquid if Heat had not become Latent.	
		°	
Water	966	966	Regnault.
Alcohol	457	735	Ure.
Ether	313	473	"
Oil of Turpentine	184	390	"
Naphtha	184	443	"

evaporate it to steam is 212 + 966 = 1178 units. Now say that the atmospheric pressure were reduced to half, or to 15 inches of mercury in the barometer, then by Table 11 water would boil at 180° instead of 212°, but the units of heat from 0° would still be 1178 as before, and as 180° only were required to raise the water to its new boiling point, the latent heat of vapourization must be 1178 − 180 = 998 units instead of 966. Again, at a pressure of 60 lbs. per square inch above the atmosphere, water must be heated to 307° before ebullition and vapourization commences, as shown by Tables 15, 71, &c., but the latent heat of vapourization will be proportionally diminished, and will in that case become 1178 − 307 = 871 units instead of 966 units.

For convenience of calculation, it is assumed in the above that water could be reduced to 0° without passing to ice, where as we have seen its specific heat is altered (3). This, of course, is fictitious; the real amount of heat required to convert a pound of ice at 0° to steam at 212° is 1304·5 units, as follows:—

			Units.
Ice at 0° to ice at 32° = 32 × ·504	=	..	16·1
Ice at 32° to water at 32° (latent)	=	..	142·4
Water at 32° to water at 212°	=	..	180·0
Water at 212° to steam at 212° (latent)	=	..	966·0
		Total	1304·5

But this difference will not affect the correctness of our rule; for instance, the amount of heat to convert water at 32° to steam is by the above investigation $180 + 966 = 1146$ units, and by the other method $1178 - 32 = 1146$ units also.

This law applies to other liquids, allowance being made for the specific heat of the particular one evaporated; thus we have

For Water	$H = 1178 - t$
„ Alcohol	$H = (908 - t) \times \cdot 622$
„ Ether	$H = (756 - t) \times \cdot 477$
„ Naphtha	$H = (730 - t) \times \cdot 434$
„ Oil of Turpentine	$H = (706 - t) \times \cdot 472$

in which H = the total heat to evaporate 1 lb. of the liquid from the temperature t to vapour or steam at any pressure.

(19.) These results, however, will be somewhat modified by the fact that both the latent and the specific heat of water vary with the temperature (3), as shown by the experiments of Regnault. The latent heat of vapourization for water will be given precisely by the rule,

$$L = 1115 \cdot 2 - (\cdot 708 \times t),$$

L being the latent heat at the temperature t. Thus, by Table 71, 50-lb. steam has a temperature of 298°, and the latent heat by the rule is $1115 \cdot 2 - (\cdot 708 \times 298) = 904 \cdot 2$ units, instead of $1178 - 298 = 880$ units as by the simple rule.

The total heat to convert a pound of water at 32° to vapour at any other temperature will be accurately given by the rule,

$$H' = 1081 \cdot 4 + (\cdot 305 \times t),$$

in which H′ = the total heat to convert water at 32° to steam or vapour at the temperature t. Thus, by Table 71, 60-lb. steam has a temperature of 307°, and the total heat from 32° is $1081 \cdot 4 + (\cdot 305 \times 307) = 1175$ units, instead of $1178 - 32 = 1146$ units as by the simple rule. Table 15 has been obtained by these rules, and should be used where great accuracy is required for scientific purposes; the simple rule is sufficiently accurate for ordinary practice.

A result of the rules in (18) is that the heat required to produce

steam is independent of the pressure of that steam; for instance, to evaporate a cubic foot of water to steam in an open vessel will require the same fuel as to convert it to steam of 50 or 100 lbs. per square inch. Col. 4 of Table 15 shows, however, that this is not strictly correct, the total heat increasing with the pressure, but the difference is very small, for even with so great a range of pressure as from 7 to 200 lbs. per square inch it amounts to only $1200 \div 1152 = 1 \cdot 04$, or 4 per cent.

TABLE 15.—Of the HEAT required to convert WATER to STEAM of different PRESSURES.

Pressure above the Atmosphere in lbs. per Square Inch.	Temperature of the Steam.	Units per lb. Water.	
		Latent Heat.	Total Heat from 32°.
	°		
7	232	950	1152
15	250	937	1157
20	259	931	1160
25	267	926	1163
30	274	920	1165
45	292	908	1171
60	307	897	1175
75	320	888	1179
100	338	876	1184
125	353	865	1189
150	366	856	1193
175	377	848	1196
200	388	840	1200
(1)	(2)	(3)	(4)

(20.) "*Expansion of Solids.*"—The expansion of solids may be estimated in two ways; by the increase in length, and by the increase in volume. Imagine a very expansible solid, such that by a given change of temperature its length was doubled, say from 1 foot to 2 feet; but obviously the breadth and height would be simultaneously doubled also, and thus if before expansion the body were a cube or $1 \times 1 \times 1 = 1$ cubic foot, it would become by expansion $2 \times 2 \times 2 = 8$ cubic feet. Estimating by the effect on the length, we should say that the comparative lengths at the two temperatures were 1 to 2, but by volume as 1 to 8, and in all cases the expansion in volume is the cube of the expansion in length.

But for the exceedingly small dilatations of solids such as are

TABLE 16.—Of the EXPANSION of BODIES by HEAT for 1° Fahrenheit, being the Mean Expansion per Degree between 32° and 212°, the Volume at 32° being 1·0.

	Expansion for 1° Fahrenheit.		No. of Authorities.
	In Length.	In Volume.	
Fire Brick	·000002349	·000007047	1
Marble (black)	·000002407	·000007420	1
White Deal	·000002556	·000007669	2
Brick, stock	·000003057	·000009170	1
Marble (Carrara)	·000003633	·000010900	1
Granite (Aberdeen grey)	·000004386	·000013157	1
*Glass tube	·000004567	·000013701	5
Platina	·000004835	·000014506	2
Slate (Penrhyn)	·000005764	·000017290	1
Cast Iron	·000006167	·000018501	3
Steel, rod	·000006441	·000019324	4
Wrought Iron	·000006689	·000020067	4
Iron Wire	·000007430	·000022290	2
Roman Cement	·000007972	·000023915	1
Copper	·000010088	·000030264	4
Brass, cast	·000010417	·000031250	1
„ plate	·000010450	·000031350	3
„ wire	·000010723	·000032170	1
Silver	·000011121	·000033364	6
Tin	·000013102	·000039307	4
Lead	·000015876	·000047628	2
Zinc, hammered	·000017268	·000051806	1
Ice, from − 17° to + 30°	·000028567	·000085700	3
Guttapercha, 25° to 60°	·000084300	·000253000	1
ABSOLUTE EXPANSION OF LIQUIDS.			
*Mercury	..	·000100540	7
†Water, 40° to 212°	..	·0002519	1
Alcohol, 30° to 100°	..	·0006455	1
„ at 32°	..	·0005830	1
Sulphuric Ether at 32°	..	·0008406	1
Sulphuret of Carbon at 32°	..	·0006330	1
Linseed Oil, 32° to 212°	..	·0004167	1
EXPANSION IN GLASS.			
Mercury	..	·000086839	..
Water, 40° to 212°	..	·0002380	1
Alcohol, 30° to 100°	..	·0006318	1
„ at 32°	..	·0005693	1
Sulphuric Ether at 32°	..	·0008269	1
Sulphuret of Carbon at 32°	..	·0006193	1
Linseed Oil, 32° to 212°	..	·0004030	1

* See Table 23. † See Table 21.

met with in experience, the expansion in volume may without sensible error be taken at three times the linear dilatation, for a cube has three dimensions, length, breadth, and height, and if each of these dimensions be increased by a very small amount, it is evident that the expansion of the cube in volume is very nearly three times the linear expansion.

Table 16 gives the expansion of bodies in length and volume: its use is very simple; thus, an iron wire 100 feet or 1200 inches long, with 55° increase in temperature, would expand $\cdot 00000743 \times 1200 \times 55 = \cdot 4904$ inch, or $\frac{1}{2}$ inch barely. Again, 10 cubic feet of linseed oil heated 240° would expand $\cdot 0004167 \times 10 \times 240 = 1 \cdot 000$ cubic foot; thus the 10 cubic feet would become by expansion 11 cubic feet, &c.

(21.) "*Contraction of Metals in Casting.*"—The contraction which metals experience in cooling down from their melting points to ordinary temperatures is very considerable, amounting to about an inch with a straight bar of cast iron 8 feet long, or with a copper bar 5 feet long. Allowance has therefore to be made for contraction in fixing the sizes of the pattern.

Table 17 gives the result of practical observations on this subject, and is very simple in application; thus a cast-iron girder 20 feet long must have a pattern $\cdot 1246 \times 20 = 2 \cdot 492$ inches longer than itself, but a pattern 20 feet long would give a casting $\cdot 1236 \times 20 = 2 \cdot 472$ inches shorter than itself.

For practical purposes $\frac{1}{8}$ of an inch to a foot for cast iron; $\frac{1}{6}$ of an inch for gun-metal; $\frac{1}{8}$ of an inch for copper; and $\frac{1}{4}$ inch for zinc may be taken as sufficient approximations.

(22.) The contraction of wheels is anomalous, as is shown by Table 18. The irregularities in the *apparent* contraction arise in part from the practice of "rapping" the pattern in the sand to make it an easy fit and enable it to be drawn out with facility. This is most influential in its results with small heavy wheels of great width of face: in some cases, and in rough hands, the casting of a small and heavy pinion may be quite the full size of the pattern. The allowance to be made is therefore not uniform, but must be fixed with judgment; in large wheels, where the effect of rapping is comparatively very small, $\frac{1}{10}$ of an inch to a foot may be taken safely. A wheel, &c., is not so free to contract as a straight bar, and in any case its contraction will be less.

TABLE 17.—Of the CONTRACTION of METALS in CASTING.

	Length of Pattern.		Contraction.			
			Total in Inches.	Per Foot		
				Of Pattern.	Of Casting.	
	ft.	in.		inches.	inches.	
Cast-iron girder	21	8¾	2$\frac{11}{16}$	·1236	·1246	
" "	16	9	2·05	·1225	·1236	
Gun-metal bar	5	4⅝	1·0	·18568	·1886	Maximum.
"	5	7$\frac{15}{16}$	·936	·1653	·1676	
"		"	·97	·1713	·1737	
"	6	0¼	1·0	·1661	·1684	
"	5	6$\frac{1}{16}$	·92	·1671	·1695	
"		"	·90	·1635	·1657	
"		"	·88	·1598	·1620	
"		"	·84	·1526	·1545	Minimum.
"		..	..	·1607	·1632	Mean of 8.
Copper and Tin, Copper 113, Tin 10	5	6$\frac{3}{16}$	·895	·1623	·1645	Maximum.
" "		"	·880	·1595	·1617	
" "		"	·880	·1595	·1617	
" "		"	·855	·1550	·1570	Minimum.
" "		"	..	·1591	·1612	Mean of 4.
Yellow Brass	2	9⅛	·5	·1811	·1839	
Copper	7	10⅞	1·54	·1948	·1980	Minimum.
"	7	5⅛	1·465	·1972	·2005	
"		"	1·465	·1972	·2005	Maximum.
"		..	..	·1964	·1996	Mean of 4.
Lead	2	0	·21	·1050	·1059	
Zinc, cast in iron mould ..	2	0$\frac{3}{16}$	·455	·2257	·2301	Minimum.
" " ..		"	·465	·2307	·2352	Maximum.
" " ..		..	..	·2282	·2326	Mean of 2.

(23.) "*Contraction of Wrought Iron.*"—When a bar of wrought iron is heated to redness and quenched in water it becomes permanently shorter than before. This fact is well known to practical men, who sometimes avail themselves of it when a wrought-iron crank, &c., has been accidentally bored out too large for its shaft; by one or more heats it may be reduced so as to be a good fit.

Table 19 gives the result of experiments on a bar of wrought iron ⅞ inch diameter and 40 inches long; it was heated to a clear orange-red in daylight, say about 2000°, and quenched in water in the usual way. The reduced results in col. 4, &c.,

TABLE 18.—Of the CONTRACTION in CASTING SPUR WHEELS in CAST IRON.

Extreme Diameter of Wheel Casting.		Pitch in Inches.	Width of Teeth in Inches.	Contraction.		
				Total in Inches.	Per Foot	
					Of Casting.	Of Pattern.
ft.	in.				inches.	inches.
10	$2\frac{3}{8}$	$3\frac{1}{2}$	12	1·08	·1059	·1040
6	$2\frac{9}{16}$	$3\frac{1}{4}$	9	·54	·0893	·0886
6	$1\frac{3}{8}$	$3\frac{1}{2}$	11	·375	·0613	·0610
5	$5\frac{9}{16}$	$3\frac{1}{2}$	11	·345	·0631	·0628
2	$11\frac{7}{8}$	$3\frac{1}{2}$	12	·11	·03896	·03884
2	$4\frac{11}{16}$	$3\frac{1}{4}$	9	·115	·0397	·0396

TABLE 19.—Of the CONTRACTION of a BAR of WROUGHT IRON 40 inches long, by repeated Heating and Quenching.

Number of Heats.	Observed Contraction in Inches.		Reduced Results.		
			Contraction per Heat.	Total Contraction.	
	Per Heat.	Total.	Inches.	Inches.	In parts of the Length.
1	·041	·041	·0400	·0400	·00100
2	·030	·071	·0340	·0740	·00185
3	·033	·104	·0260	·1000	·00250
4	·015	·119	·0230	·1230	·00307
5	·030	·149	·0225	·1455	·00364
6	·020	·169	·0220	·1675	·00419
7	·019	·188	·0215	·1890	·00472
8	·009	·197	·0210	·2100	·00525
9	·021	·218	·0205	·2305	·00576
10	·030	·248	·0200	·2505	·00626
11	·021	·269	·0195	·2700	·00675
12	·021	·290	·0190	·2890	·00722
13	·025	·315	·0185	·3075	·00769
14	·008	·323	·0180	·3255	·00814
15	·019	·342	·0175	·3430	·00857
16	·027	·369	·0170	·3600	·00900
17	·013	·382	·0165	·3765	·00941
18	·009	·391	·0160	·3925	·00981
19	·015	·406	·0155	·4080	·01020
20	·007	·413	·0150	·4230	·01057
(1)	(2)	(3)	(4)	(5)	(6)

were obtained by making a diagram of the experimental ones in col. 2 and drawing a curve, by which the irregularities of the experiments were eliminated. It will be observed by col. 4 that the contraction per heat is continuously reduced with each successive heat, becoming with the tenth heat about half that with the first. A bar 95 inches long would be reduced 1 inch, or to 94 inches, by twenty heats.

(24.) Table 20 gives the result of experiments made at Swindon on a flanged wheel-tire 7 feet diameter, or 22 feet circumference, $5\frac{1}{2}$ inches wide and $2\frac{1}{2}$ inches thick. The contractions are very much greater than those in Table 19; the cause does not appear, perhaps the temperature was much higher, but that would not be sufficient to explain the difference in the results.

TABLE 20.—Of the CONTRACTION of a WHEEL TIRE 7 feet diameter, by repeated Heating and Quenching.

Number of Heats.	Observed Contractions.		
	Per Heat.	Total.	In parts of the Length.
	inches.	inches.	
1	$\frac{5}{8}$	$\frac{5}{8}$	·00237
2	$\frac{11}{16}$	$1\frac{5}{16}$	·00500
3	$\frac{7}{16}$	$1\frac{3}{4}$	·00667
4	$\frac{5}{8}$	$2\frac{3}{8}$	·00900
5	$\frac{3}{8}$	$2\frac{3}{4}$	·01042
6	$\frac{1}{4}$	3	·01136
7	$\frac{1}{4}$	$3\frac{1}{4}$	·01231
8	$\frac{3}{8}$	$3\frac{5}{8}$	·01373
9	$\frac{1}{4}$	$3\frac{7}{8}$	·01468
10	$\frac{1}{2}$	$4\frac{3}{8}$	·01657
11	$\frac{5}{8}$	5	·01900
12	$\frac{1}{4}$	$5\frac{1}{4}$	·02000
13	$\frac{1}{8}$	$5\frac{3}{8}$	·02030
(1)	(2)	(3)	(4)

(25.) "*Expansion of Liquids.*"—The expansion of liquids must be estimated by the increase in *volume*. Referring to the former illustration (20), we may suppose that a cubic foot of the liquid is contained in a vessel that *does not itself expand with heat*, but of such a *height* as to allow the liquid to expand in that direction only. When, by expansion, the cubic foot of liquid becomes 8 cubic feet, it is obvious that the vessel, whose

length and breadth is fixed, must be 8 feet high to hold the expanded liquid, and thus the *linear* dilatation is in fact the actual expansion in *volume*.

But if the vessel is itself expansible, the observed expansions are apparent only, not real and absolute, being in fact the *difference* between the expansion of the liquid and that of the vessel containing it. Thus from Table 16 the expansion of glass in volume is ·000013701, and the *absolute* expansion of mercury is ·00010054, the *apparent* expansion of mercury in a glass vessel (such as a thermometer bulb, &c.) will therefore be ·00010054 − ·000013701 = ·000086839, as per Table 16.

The expansion of water is exceptional and anomalous. It attains a minimum volume and a maximum density at 39°·2, say 40°, and a departure from that temperature, in either direction, is accompanied by expansion, so that 8° or 10° of cold produces about the same amount of expansion as 8° or 10° of heat. This is shown by Table 21, which is calculated by Tredgold's rule,

$$\frac{5}{3}\log.(t-40)+\bar{6}\cdot910909 = \text{the log. of the expansion.}$$

Thus at 212° we have 212−40 = 172°, the logarithm of which is 2·235528.

Then, from 40° to 212°, the expansion becomes

$$\begin{array}{rr}
 & \text{Log.} \\
212-40=172^\circ = & 2\cdot235528 \\
 & 5 \\
\hline
3) & 11\cdot177640 \\
\hline
 & 3\cdot725880 \\
 & \bar{6}\cdot910909 \\
\hline
 & \bar{2}\cdot636789 = \cdot04333;\ \text{the expansion.}
\end{array}$$

Hence the volume at 40° being 1·0, that at 212° is 1·04333, &c., as in the table.

In the act of freezing, water expands very considerably a

cubic foot of water at 32° weighs 62·38 lbs., but a cubic foot of ice at 32°, only 57·96 lbs. by Table 37, hence the floating power is 62·38 − 57·96 = 4·42 lbs. per cubic foot, and a man weighing 150 lbs. would be carried by 150 ÷ 4·42 = 34 cubic feet of ice. In icebergs, &c., the part submerged is about 57·96 ÷ 4·42 = thirteen times the size of the part above water.

TABLE 21.—Of the VOLUME, SPECIFIC GRAVITY, EXPANSION, and WEIGHT of WATER at different Temperatures.

Temp. Fahr.	Volume.	Specific Gravity.	Weight of a Cubic Foot in Pounds.	Expansion for 1° between the different Temperatures.
°				
20	1·0012000	·99880	62·33	·0000822
30	1·0003780	·99962	62·38	·0000378
40	1·0000000	1·00000	62·408	·0000129
42	1·0000258	·99997	62·406	·0000486
52	1·0005123	·99950	62·377	·0000895
62	1·0014070	·9986	62·321	·0001216
72	1·002627	·9974	62·25	·0001516
82	1·004143	·9959	62·15	·0001758
92	1·005901	·9941	62·04	·0002010
102	1·007911	·9921	61·92	·0002239
112	1·010150	·9900	61·78	·000246
122	1·01261	·9875	61·63	·000266
132	1·01527	·9850	61·47	·000287
142	1·01814	·9822	61·30	·000306
152	1·02120	·9792	61·11	·000323
162	1·02443	·9761	60·92	·000345
172	1·02788	·9729	60·72	·000360
182	1·03148	·9695	60·50	·000378
192	1·03526	·9659	60·28	·000396
202	1·03922	·9622	60·05	·000411
212	1·04333	·9585	59·82	·000434
230	1·05115	·9513	59·37	·000464
250	1·06043	·9430	58·85	·000498
275	1·07289	·9321	58·17	·000562
300	1·08693	·9200	57·42	·000573
350	1·11560	·8963	55·94	·000656
400	1·14840	·8708	54·34	·000718
450	1·18430	·8444	52·70	·000780
500	1·22330	·8175	51·02	·000866
600	1·30990	·7634	47·64	

(26.) "*Variation in Expansion.*"—The expansion of solids, and liquids are not equal for equal increments of heat, but increase

with the temperature, as shown by the experiments of Dulong and Regnault in Table 22. The rate of expansion being variable with the temperature it is necessary to distinguish between expansions *at* a given temperature and those *between* two given temperatures. For example, imagine a body which *at* 32° expands ·01 for 1°, and at 212°, ·02 for 1°; then supposing the rate of expansion to increase in arithmetical progression, the *mean* expansion between 32° and 212° would be (·01 + ·02) ÷ 2 = ·015, which is in fact the true rate of expansion *at* the mean temperature (32 + 212) ÷ 2 = 122°. Then between 32° and 120° it would be (·01 + ·015) ÷ 2 = ·0125, and between 120° and 212° = (·015 + ·02) ÷ 2 = ·0175. Table 16 gives for the most part the expansions between 32° and 212°; where otherwise specified, the meaning will be understood from the explanation now given.

The expansions of mercury and glass (being the materials of which ordinary thermometers are made) are very important in the interests of science. They have been carefully ascertained by Regnault, and the results are given in Table 23, which has been calculated from his experiments.

TABLE 22.—Of the VARIATION in the EXPANSIONS of BODIES at different Temperatures, from the Experiments of DULONG and REGNAULT.

	32° to 212°.	32° to 392°.	32° to 572°.	Authority.
	Linear Expansion for 1° Fahr.			
Platina	·000004912	..	·000005101	Dulong.
Iron	·000006567	..	·000008158	"
Copper	·000009545	..	·000010462	"
Glass	·000004785	·000005125	·000005616	"
"	·000005113	·000005386	·000005660	Regnault.
	Expansion in Volume for 1° Fahr.			
Mercury ..	·0001001	·00010241	·00010482	Dulong.
" ..	·00010085	·000102252	·000103653	Regnault.

TABLE 23.—Of the ABSOLUTE EXPANSION of MERCURY and of COMMON GLASS, calculated from REGNAULT'S Experiments.

Temperature by an Air-Thermometer.	Volume at the given Temperature.	Expansion in Volume for 1° Fahr.	
		Between 32° and the given Temperature.	At the given Temperature.
		MERCURY.	
° 32	1·000000	..	·000099472
122	1·009013	·000100144	·000100844
212	1·018153	·000100850	·000102250
302	1·027419	·000101552	·000103650
392	1·036811	·000102252	·000105050
482	1·046329	·000102953	·000106450
572	1·055973	·000103653	·000107850
662	1·065743	·000104354	·000109255

GLASS.

Temperature by an Air-Thermometer.	Volume at the given Temperature.	Mean Expansion for 1° Fahr. between 32° and the given Temperature.	
		In Volume.	In Length.
32	1·000000	..	..
122	1·001343	·000014928	·000004976
212	1·002761	·000015338	·000005113
302	1·004252	·000015750	·000005250
392	1·005817	·000016159	·000005386
482	1·007416	·000016569	·000005523
572	1·009169	·000016979	·000005660
662	1·010958	·000017394	·000005798

(27.) "*Expansion of Gases.*"—It has been found by experiment that all the gases, dry air, and even vapours, *out of contact with their generating fluids*, expand alike or very nearly so. Fig. 85½ represents the condition of air at the surface of the earth: let A be an open-topped vessel, 1 foot square in plan, fitted with a frictionless piston B, and let C be a column of mercury 29·92 inches high, which presses on the piston and compresses the air beneath it into 1 cubic foot, as in the figure; and let the whole apparatus be placed in a vacuum surrounding

it on all sides. Say that the air in D is at 32°, and that by applying heat it is expanded and the piston with its load is raised: Regnault found that by heating from 32° to 212° or 180°, the piston in our case would be raised ·367 foot, or to E, and the cubic foot of air becomes 1·367 cubic foot: we have here supposed that the capacity of the vessel itself is not affected by heat. Now if we had taken 180° away from the air, thus reducing the temperature to −148° below zero of Fahrenheit, the piston would have descended the same distance, ·367 foot, or to F; another reduction of 180°, or to −328°, lowers the piston to G; and to cause it to descend to H, or to reduce the volume of the air to *nothing* theoretically, we should require $\frac{180}{\cdot 367} = 490^\circ\cdot 4$ below 32°, or $490\cdot 4 - 32 = 458\cdot 4$ below Fahrenheit's zero. This temperature is termed *absolute zero*, and it will be evident that the volume of air at all temperatures is proportional to the distance of its temperature from $-458^\circ\cdot 4$. Hence the general formula becomes

$$v = V \times \frac{458\cdot 4 + t}{458\cdot 4 + T},$$

in which V = volume of gas, &c., at the temperature T,
v = „ „ at the new temperature t.

Thus air whose volume at 32° = 1·000, will have at 2500° a volume of $1\cdot 000 \times \frac{458\cdot 4 + 2500}{458\cdot 4 + 32} = 6\cdot 032$. Table 24 has been calculated by this rule, and Table 25 gives a comparison of its results with the experiments of Dulong and Petit.

28.) The experiments of Regnault have shown that although practically all the gases and *dry* vapours expand alike, and equally for equal increments of heat, yet that both those statements are not rigorously true; but that air and all gases except hydrogen have coefficients of dilatation, which increase slightly with their density, but approach more nearly to equality as their pressures become feeble. Table 26 shows that with air and carbonic acid gas the rate of expansion increases slightly with the pressure, which varying in the case of air

TABLE 24.—Of the VOLUME and WEIGHT of DRY AIR at different Temperatures under a constant Atmospheric Pressure of 29·92 inches of Mercury in the Barometer, the Volume at 32° being 1.

Temperature.	Volume.	Weight of a Cubic Foot in Pounds.	Temperature.	Volume.	Weight of a Cubic Foot in Pounds.
0°	·935	·0864	550°	2·056	·0384
12	·960	·0842	600	2·158	·0376
22	·980	·0824	650	2·260	·0357
32	1·000	·0807~~4~~5	700	2·362	·0338
42	1·020	·0791	750	2·464	·0328
52	1·041	·0776	800	2·566	·0315
62	1·061	·0761	850	2·668	·0303
72	1·082	·0747	900	2·770	·0292
82	1·102	·0733	950	2·872	·0281
92	1·122	·0720	1000	2·974	·0268
102	1·143	·0707	1100	3·177	·0254
112	1·163	·0694	1200	3·381	·0239
122	1·184	·0682	1300	3·585	·0225
132	1·204	·0671	1400	3·789	·0213
142	1·224	·0660	1500	3·993	·0202
152	1·245	·0649	1600	4·197	·0192
162	1·265	·0638	1700	4·401	·0183
172	1·285	·0628	1800	4·605	·0175
182	1·306	·0618	1900	4·809	·0168
192	1·326	·0609	2000	5·012	·0161
202	1·347	·0600	2100	5·216	·0155
212	1·367 5	·0591	2200	5·420	·0149
230	1·404	·0575	2300	5·624	·0142
250	1·444	·0559	2400	5·828	·0138
275	1·495	·0540	2500	6·032	·0133
300	1·546	·0522	2600	6·236	·0130
325	1·597	·0506	2700	6·440	·0125
350	1·648	·0490	2800	6·644	·0121
375	1·689	·0477	2900	6·847	·0118
400	1·750	·0461	3000	7·051	·0114
450	1·852	·0436	3100	7·255	·0111
500	1·954	·0413	3200	7·459	·0108

TABLE 25.—Of the EXPANSION of DRY AIR by HEAT.

Temperature.	Volume by Experiment. Dulong and Petit.	Volume by Calculation.	Temperature.	Volume by Experiment. Dulong and Petit.	Volume by Calculation.
−32·8°	·8650	·8678	392°	1·7389	1·734
+32	1·0000	1·0000	482	1·9198	1·918
212	1·3750	1·367	572	2·0976	2·101
302	1·5576	1·551	680	2·3125	2·322

$4 \cdot 81 \div \cdot 1441 = 33 \cdot 4$ to 1 causes the zero to vary from $-461 \cdot 4$ to $-453 \cdot 1$. Table 27 gives the exact rate of expansion for various gases at atmospheric pressure, and also shows that it varies slightly according as the volume or the pressure is taken as constant. It is somewhat anomalous to give a coefficient of *expansion* with *constant volume*, but the meaning is that the constants in cols. 1 and 2 govern the volume when the pressure is constant, and those in cols. 3 and 4 govern the pressure when the volume is constant.

For scientific purposes, therefore, instead of the coefficient 458·4 given by our rule (27), those given by col. 6 of Table 26, or by cols. 2 and 4 in Table 27 should be used. For *dry* or superheated steam Messrs. Fairbairn and Tate's experiments give the zero at − 458·71, which is almost precisely that given by the rule for air (27).

TABLE 26.—Of the VARIATION with different PRESSURES in the RATE of EXPANSION of DRY AIR and CARBONIC ACID GAS by HEAT, the Volume being constant. From the Experiments of REGNAULT.

Total Pressure by a Column of Mercury at 32°.		Ratio of Densities at 32°.	$\frac{P}{p}$	Reduced Coefficient of Expansion from 32° to 212°.	Absolute Zero with Volume Constant.
Air at 32°.	Air at 212°.				
p inches.	P inches.				
ATMOSPHERIC AIR.					
					c
4·3197	5·8665	·1444	1·3581	·36482	− 461·4
6·8645	9·3374	·2294	1·3602	·36513	− 461·0
10·4748	15·5539	·3501	1·4845?	·36542	− 460·6
14·7507	20·0925	·4930	1·3621	·36586	− 460·0
29·9221	40·8885	1·0000	1·3665	·36650	− 459·1
66·0786	90·0024	2·2084	1·3621	·36760	− 457·7
84·6164	115·1194	2·8213	1·3649	·36894	− 455·9
143·9194	196·5386	4·8100	1·3653	·37092	− 453·1
CARBONIC ACID GAS.					
29·8609	40·7298	1·0000	1·36397	·36856	− 456·4
35·4759	47·6317	1·1879	1·35434	·36942	− 455·3
68·6113	94·0045	2·2976	1·37089	·37523	− 448·8
141·3017	187·3630	4·7318	1·32040	·38598	− 434·3
(1)	(2)	(3)	(4)	(5)	(6)

TABLE 27.—Of the ABSOLUTE EXPANSION of GASES from 32° to 212°, by the Experiments of REGNAULT.

	Pressure Constant.		Volume Constant.	
	Expansion.	Absolute Zero.	Expansion.	Absolute Zero.
Air, Atmospheric	·3670	− 458·4	·3665	− 459·1
Hydrogen	·3661	− 459·7	·3667	− 458·9
Nitrogen	..	..	·3668	− 458·7
Carbonic Acid	·3710	− 453·2	·3688	− 456·1
Carbonic Oxide	·3669	− 458·6	·3667	− 458·9
Sulphurous Acid	·3903	− 429·2	·3845	− 436·2
	(1)	(2)	(3)	(4)

(29.) "*Law of Marriotte.*"—When the pressure is not constant the volume may be calculated by the law of Marriotte, namely, that *the volume of any gas varies in the inverse ratio of the pressure —the temperature remaining constant.* The pressure here meant is the total pressure above a vacuum. Thus, a cubic foot of air in ordinary cases has the pressure of the atmosphere upon it to begin with, say 15 lbs. (nearly) per square inch, and its volume under a pressure of say 45 lbs. per square inch above the atmosphere will be $1 \times 15 \div (15 + 45) = \cdot 25$ cubic foot. Practical men have generally to deal with pressures above the atmosphere, and are apt to forget to allow for the same, and thus make serious errors.

(30.) It appears from the experiments of Regnault that this old law of Marriotte is only approximately true, as shown by Table 28; thus by increasing the density from 1 to 20 we should with any of the gases have to increase the pressure from 1 to 20 also, but the table shows that with the exception of hydrogen where the pressure is greater, the other gases require a rather less pressure than is due by the law of Marriotte. With the exception of carbonic acid, however, the differences are small, and may be neglected in practice.

(31.) When there is a change both in temperature and pressure, the rule becomes

$$v = V \times \frac{P}{p} \times \frac{458 \cdot 4 + t}{458 \cdot 4 + T},$$

TABLE 28.—Of the DENSITY and CORRESPONDING PRESSURE of AIR and GASES. REGNAULT.

Density.	Atmospheric Air.	Nitrogen.	Carbonic Acid.	Hydrogen.
1	1·000000	1·000000	1·00000	1·000000
5	4·979440	4·986760	4·82880	5·011615
10	9·916220	9·943590	9·22620	10·056070
15	14·824845	14·875770	13·18695	15·139650
20	19·719880	19·788580	16·70540	20·268720

in which V, P and T are the Volume, Pressure, and Temperature in one case, and v, p and t are the Volume, Pressure, and Temperature in another case. Thus 10 cubic feet of air at the ordinary atmospheric pressure, say 15 lbs. per square inch above a vacuum, and temperature 60°, would if heated to 200° and under a pressure of 40 lbs. above atmosphere, or $15 + 40 = 55$ lbs. above a vacuum, become $10 \times \frac{15}{55} \times \frac{458 \cdot 4 + 200}{458 \cdot 4 + 60} = 3 \cdot 7$ cubic feet.

(32.) "*Expansion of Moist Air.*"—When water or other liquid is present in the air or gas, another element becomes necessary in the calculation of its bulk at different temperatures, namely, the elastic force of the vapour at the given temperature:—the rule then becomes

$$v = \text{V} \times \frac{p + f}{\text{P} + \text{F}} \times \frac{458 \cdot 4 + t}{458 \cdot 4 + \text{T}},$$

in which V, P, T and F are the Volume, Pressure, Temperature, and elastic Force of vapour of the given liquid in one case, and v, p, t, and f the corresponding Volume, Pressure, Temperature, and elastic Force of vapour of the given liquid in another case. The principle of this rule will be best understood by reasoning out an example, taking the elastic force of water from Table 67.

(33.) Imagine a vessel containing 1000 cubic feet of air at 0°, saturated with vapour, and having a barometer enclosed indicating say 32·18 inches of mercury as the pressure of the mixture of air and vapour. Now it is an axiom that the pressure of such a mixture is the sum of the separate pressures or elasticities of the air and the vapours; and as the elastic force of

vapour at 0° is ·044, it follows that the pressure of the air alone is in our case 32·18 − ·044 = 32·136 inches. If the mixture be heated to say 112°, it will behave exactly as *dry* air or gas if no water be present to supply vapour; and the pressure remaining the same, the volume would become $1000 \times \frac{458 \cdot 4 + 112}{458 \cdot 4 + 0°} = 1244$ cubic feet. If now a little water be added to saturate the heated air, the tension of vapour at 112° being 2·731, the pressure would in a closed vessel be increased to 32·136 + 2·731 = 34·867; and if the vessel be enlarged until that pressure be reduced to its normal state of 32·18 inches, its capacity would become $1244 \times \frac{34 \cdot 867}{32 \cdot 18} = 1348$ cubic feet. Putting this in the form of the formula already given, we have $1000 \times \frac{32 \cdot 136 + 2 \cdot 731}{32 \cdot 136 + \cdot 044} \times \frac{458 \cdot 4 + 112}{458 \cdot 4 + 0} = 1348$ cubic feet, as before. General Roy made some experiments on the expansion of moist air at a pressure of 32·18 inches. His results are given in Table 29; col. 4 is calculated by the formula.

If the vapour present is other than that of water, of course the elastic force of that particular vapour must be taken, from Table 72.

TABLE 29.—Of the EXPANSION of MOIST AIR, at a Pressure of 32·18 in. of the Barometer, from the Experiments of GENERAL ROY.

Temperature.	Elastic Force of Vapour, Inches of Mercury.	Volume by Experiment.	Volume by Calculation.
°			
0	·044	1000·00	1000·0
32	·181	1071·29	1074·5
52	·388	1123·05	1125·1
72	·785	1182·50	1183·7
92	1·501	1255·14	1255·0
112	2·733	1353·75	1348·3
132	4·752	1491·06	1476·4
152	7·931	1689·00	1657·0
172	12·752	1929·78	1918·3
192	19·837	2287·44	2291·0
212	30·000	2672·00	2824·0

(34.) "*Thermometers.*"—The expansion of bodies by heat has afforded the most convenient method of measuring temperatures. In the common thermometers mercury is used for medium temperatures, say from 0° to 600°; for lower temperatures alcohol is used, because it always remains fluid, even with the greatest cold which can be produced by artificial means; for high temperatures metals are commonly used. There is an imperfection in all these bodies as measurers of heat, for as we have seen (26) their expansions are not equable for equal increments of heat; but in the case of the mercurial thermometer it most fortunately happens that the variations of expansion in glass and mercury almost exactly compensate each other, so that mercury in glass has an expansion nearly equable at medium temperatures. Table 30, calculated from the refined experiments of Regnault, gives the error of the mercurial thermometer in common glass, such as is ordinarily used for thermometers, and it shows that, for temperatures under 540°, the error is less than 1° Fahr.; above that temperature the error becomes rapidly greater, and amounts to 7°·2 at 662°. The amount of error seems to vary greatly with the kind of glass employed: with fine crystal glass the error at 662° was found by Regnault to be as much as 19°.

TABLE 30.—Of the ERROR of the COMMON MERCURIAL THERMOMETER in GLASS TUBE, from the Experiments of REGNAULT.

Temperature by an Air-Thermometer.	Error of Mercurial Thermometer in Degrees Fahr.	Temperature by an Air-Thermometer.	Error of Mercurial Thermometer in Degrees Fahr.	Temperature by an Air-Thermometer.	Error of Mercurial Thermometer in Degrees Fahr.
°		°		°	
212	·000	374	−·630	536	+ ·936
230	−·036	392	−·550	554	+1·440
248	−·090	410	−·450	572	+1·944
266	−·162	428	−·360	590	+2·610
284	−·270	446	−·270	608	+3·240
302	−·360	464	−·180	626	+4·320
320	−·468	482	+·090	644	+5·400
338	−·576	500	+·360	662	+7·200
356	−·666	518	+·684		

(35.) "*Air-Thermometers.*"—Air offers the great advantage as a measurer of heat, that its expansions are nearly equal for

equal increments of heat (27). This property renders it particularly valuable for very high temperatures where the expansions of solids become very irregular. Air-thermometers, however, are not adapted for ordinary use, and for practical purposes less perfect but more convenient means have to be adopted for measuring high temperatures.

(36.) "*Immersion Thermometer.*"—A simple method of estimating high temperatures is heating a mass of wrought iron to the unknown temperature, immersing it in a known weight of water, and observing the increase of temperature produced.

Thus, let 7 lbs. of wrought iron, which has been heated to the unknown temperature, be plunged into 10 lbs. of water at 60°, heating it thereby to 180°. Then $(180 - 60) \times 10 = 1200$ units of heat have been given out by the 7 lbs. of iron, or $1200 \div 7 = 171 \cdot 43$ units per pound, and the specific heat of wrought iron being by Table 1 equal to one-ninth that of water, it must have been cooled $171 \cdot 43 \times 9 = 1543°$; the unknown temperature must therefore have been $1543 + 180 = 1723°$.

Putting this into the form of a rule:—

Let t = the temperature of the water before immersion.
T = " " after "
T' = the unknown temperature required.
W = the weight of the water in lbs., &c.
w = the weight of the wrought iron in lbs., &c.

$$\text{Then } T' = \Big((T - t) \times W \times 9 \div w\Big) + T.$$

Thus in our case $\Big((180 - 60) \times 10 \times 9 \div 7\Big) + 180 = 1723°$ the temperature required. There are several sources of inaccuracy in this method, namely, the variable specific heat of iron (2) and water (3), &c.; notwithstanding which, it is perhaps the best practical method we have for high temperatures beyond the range of the mercurial thermometer, or say 600° Fahr.

(37.) "*Colour Thermometer.*"—The colours which polished steel assumes when heated, have been used by workmen from time immemorial as a guide for temperature in tempering steel instruments. After heating to redness and quenching in water, the steel becomes exceedingly hard and brittle: it is then roughly

polished and again heated until it takes the colour known by experience to be necessary to obtain the particular temper required, which varies with the nature of the work which the tool is to be used for. As the temperature rises the colour changes successively from white to yellow, brown, red, purple, blue, dark blue, green, and finally to dark grey. Table 31 gives the colours with the respective temperatures, &c. Temperatures ranging between 400° and 600° might be conveniently estimated by this method; a small piece of thin steel re-polished for each observation would indicate with sufficient precision the temperature of an oven, &c., &c. An ordinary oven requires about 450° of heat, and should give to steel a straw yellow.

Pouillet states that the temperature of furnaces, &c., may be estimated with considerable accuracy by the colour of the fire, and that with a little practice the error at very high temperatures will not exceed 90° or 100°. Table 32 gives the result of his observations with an air-thermometer.

TABLE 31.—Of the COLOURS of POLISHED STEEL at different Temperatures.

Kind of Tools, &c.	Temperature.
	°
Very faint yellow for lancets	420
Pale straw yellow for razors, &c.	440
Orange for penknives and chisels	470
Brown for scissors, &c.	490
Red for carpenters' tools	510
Purple for watch-springs	530
Bright blue for lock-springs	550
Full blue for fine saws and needles	560
Dark blue for common saws	600
Greenish	630
Grey	750

(38.) "*Scales of the Thermometer.*"—The *scale* of the thermometer is arbitrary; in this country the scale of Fahrenheit is used, in which two standard points are fixed, the lower one is obtained by plunging the thermometer into melting ice (13), and is marked 32°: the upper one is the temperature of *steam* with a pressure of 29·905 inches of mercury at 32° in the barometer (16), and is marked 212°. Thus the difference between the two

TABLE 32.—TEMPERATURES corresponding to various DEGREES of LIGHT in HEATED METAL, FURNACES, &c. M. POUILLET.

	Fahr.
	°
Red, just visible	977
„ dull	1290
„ cherry, dull	1470
„ „ full	1650
„ „ clear	1830
Orange, deep	2010
„ clear	2190
White heat	2370
„ bright	2550
„ dazzling	2730

points is divided into $212 - 32 = 180°$, which process continued upwards and downwards gives the whole scale of the instrument. The whole length of the stem up to 212° should be heated by the steam, for obviously if the bulb only were so heated while the long column of mercury in the stem was at the low temperature of the ambient air, a considerable error would ensue.

All thermometers intended for scientific purposes should be so graduated, and so used in practice, or otherwise the maker should indicate on each instrument the height to which it should be immersed in the liquid whose temperature was being taken. Care should also be taken to ascertain that the bore is equable throughout its length, by observing the length of a short column of mercury in different parts of the tube.

On the Continent the Centigrade scale is commonly used; here the same standard points are used, but the lower one is marked 0° and the upper 100°, hence the distance between the two points is divided into 100°. The readings of the two scales are easily convertible by the rules

$$F = (C \times 1\cdot8) + 32 \qquad \text{and } C = (F - 32) \div 1\cdot8,$$

in which F and C represent degrees in the respective scales Fahrenheit and Centigrade: thus, 75° Cent. is $= (75 \times 1\cdot8) + 32 = 167°$ Fahr.; and 256° Fahr. is $= (256 - 32) \div 1\cdot8 = 124\cdot4$ Cent., &c. Plate 12 gives a direct comparison of these scales for a great range of temperature.

(39.) "*The Position of Thermometers.*"—A thermometer freely exposed in the open air is subjected to four distinct and sometimes contrary influences. There is solar radiation; radiation from the cold sky; radiation from the earth; and contact of the ambient air with the bulb. In such a case the thermometer would not show the temperature of either the sun, sky, earth, or air, but a combined result of the whole in unknown proportions. For the purposes of science we require to measure at least three of these influences separately, namely, the temperature of the air, of solar radiation, and sky radiation, and we may indicate briefly the arrangements necessary for the purpose.

(40.) "*Temperature of the Air.*"—The thermometer should be completely screened from radiation of heat or cold in *all* directions, but a screen of single thickness will not perfectly answer that purpose, because it will absorb the radiant heat and then radiate that heat to the bulb. The screen should be double or treble, with a space of 1 inch at least between each, completely open at the ends to permit a current of air: the effect of the outer case being heated would then be to heat the air in the included space, which being heated would become lighter, and ascending would carry off the heat and so keep the inner case cool.

"*Solar Radiation.*"—A thermometer with a blackened bulb freely exposed to sun and air will show an excess of temperature over the air to an extent which varies with the seasons, being on an average 4° in January and 40° in June, but sometimes as much as 12° and 65° respectively. The mid-day temperature in the sun will on an average range from 47° in January to 111° in June, but occasionally rising to 65° and 155° respectively. See Table 96.

But in such a case, part of the heat received from the sun is lost by contact of cool air, the amount of cooling varying with the temperature of the air. This loss may be prevented by enclosing the thermometer in an *exhausted* glass vessel, when on an average 20° to 30° and sometimes 40° higher temperature may be attained, and the irregularities due to the varying temperature of the air are eliminated.

(41.) "*Sky Radiation.*"—The temperature of celestial space is as low as − 224° Fahr. according to Pouillet, and if there

were no atmosphere a thermometer screened from radiation from the earth would indicate that temperature constantly. As it is, the depression does not often exceed 10° to 15° below the temperature of the air; but this is sufficient to reduce the temperature occasionally to 32°, or the freezing point in every month of the year except July and August. The daily range of the same thermometer (40) thus exposed alternately to solar and sky radiation will vary from 31° in winter to 103° in June.

(42.) "*Temperature of the Air of the Globe.*"—The temperature of the air varies not only with the geographical position, but also with the height above the sea level, and with local circumstances, so that observation alone can determine the mean temperature of any place. Table 33 gives the mean temperature of every week in the year at Greenwich, deduced from Mr. Glaisher's observations, and Table 34 gives a summary of the temperatures of forty-four principal places in all parts of the globe calculated from the table of M. Mahlman.

The influence of elevation above the sea is very considerable, but seems to vary with the climate, the season of the year, and the contour of the ground. Where the slope is gradual, the cold produced is 1° Fahr. for about 430 feet; on steep mountain slopes about 355 feet, and in balloon ascents about 330 feet.

TABLE 33.—Of the MEAN TEMPERATURE of every WEEK and MONTH in the YEAR at GREENWICH: from Fifty Years' Observations.

Months.	Mean of Month.	1st Week.	2nd Week.	3rd Week.	4th Week.
	°	°	°	°	°
January	36·5	36·2	(35·6)	36·2	37·9
February	38·4	38·0	38·1	38·3	39·3
March	41·0	40·0	40·6	41·8	42·2
April	46·0	44·7	45·2	46·3	47·8
May	52·6	51·1	51·5	53·1	54·6
June	58·8	56·9	58·0	59·5	60·6
July	61·7	61·5	61·6	61·7	61·9
August	61·4	(62·2)	61·7	61·4	60·1
September	56·7	58·3	57·4	56·1	54·9
October	50·0	52·9	51·2	49·3	47·7
November	43·3	45·8	44·0	42·3	41·2
December	39·3	40·5	40·1	39·6	37·0

TABLE 34.—Of the MEAN TEMPERATURE of the AIR, in various parts of the Globe, at different Seasons of the Year.

	Height above the Sea in feet.	Mean Temperature.				
		Year.	Spring.	Summer.	Autumn.	Winter.
		°	°	°	°	°
Irkutsk, Siberia	..	14·5	17·1	63·0	20·1	−38
Nain, Labrador	..	25·5	21·6	45·7	36·0	− 1·3
St. Bernard, Alps ..	8180	30·2	28·4	43·0	31·3	18·0
St. Gothard, Alps ..	6873	30·6	27·1	44·1	32·0	18·3
Petersburgh	..	38·3	35·1	60·3	40·5	16·7
Moscow	480	38·5	43·3	62·6	34·9	13·5
Christiania	..	41·7	39·2	59·5	42·4	25·2
Stockholm	134	42·1	38·3	61·0	43·7	25·5
Montreal	..	43·7	44·2	69·1	47·1	17·5
Warsaw	397	45·5	44·6	63·5	46·4	27·5
Berne	1918	46·0	45·8	60·4	47·3	30·4
Stromness (Orkney) ..	..	46·4	43·7	54·5	48·2	39·2
Copenhagen	..	46·8	43·7	63·0	48·7	31·3
Dresden	397	47·3	47·1	62·9	47·1	31·3
Edinburgh	288	47·5	45·7	57·9	48·0	38·5
Berlin	128	47·5	46·4	63·1	47·8	30·6
Nicolaief	..	48·7	49·3	71·2	50·0	25·9
Greenwich (Glaisher)..	..	49·2	46·9	60·8	50·2	38·9
Vienna	512	50·2	50·9	68·5	50·9	33·4
Paris	210	51·4	50·5	64·6	52·2	37·9
Penzance	..	52·0	49·8	61·7	53·8	43·9
Hobart Town	..	52·3	52·9	63·1	51·6	42·1
Turin	915	53·1	53·1	71·6	53·8	33·4
Trieste	288	55·8	53·8	71·5	56·7	39·4
Constantinople	..	56·7	51·8	73·4	60·4	40·6
Montpellier	..	57·4	56·8	75·9	61·0	44·5
Madrid	2175	57·6	57·6	74·1	56·7	42·1
Rome	174	59·7	57·4	73·2	61·7	46·6
Nice	..	60·1	55·9	72·5	63·0	48·7
Quito	9560	60·1	60·3	60·1	63·5	59·7
Naples	180	61·5	59·4	74·8	62·2	49·6
Lisbon	236	61·5	59·9	71·1	62·6	52·3
Buenos Ayres	..	62·5	59·4	73·0	64·6	52·5
Palermo	180	63·0	59·0	74·3	66·2	52·5
Algiers	..	64·0	63·0	74·5	70·5	54·0
Paramatta (Australia)	..	64·6	66·6	73·9	64·8	54·5
Madeira (Funchal) ..	..	65·7	63·5	70·0	67·6	61·3
Cape of Good Hope ..	..	66·4	65·5	74·1	66·9	58·6
Canton	..	69·8	69·8	82·0	72·9	54·8
Cairo	..	72·3	71·6	84·6	74·3	58·5
Ceylon (Candy)	1683	72·9	74·3	73·0	72·3	72·1
Rio Janeiro	..	73·6	72·5	79·0	74·5	68·5
Calcutta	..	78·4	82·6	83·3	80·0	67·8
Jamaica	..	79·0	78·3	81·3	80·0	76·3

NOTE.—The Seasons in this Table are (for the Northern Hemisphere): Spring—March April, and May; Summer—June, July, and August; Autumn—September, October, and November; and Winter—December, January, and February.

(43.) "*Internal Heat of the Globe.*"—The temperature of the *surface* of the ground follows pretty closely that of the air, which of course varies with the season, but as we descend those variations diminish until at a certain depth we come to a stratum whose temperature is invariable throughout the year, and is equal to the *mean* temperature of the air at that place. The depth of this stratum varies with the conducting power of the soil, and with the variableness of the surface climate: in tropical America it occurs at about 2 feet; in this country from 20 to 35 feet.

Between the surface and this stratum of invariable temperature we have a series of conditions approaching to the variability of the surface or the fixedness of the lower stratum, according to the depth. Thus Mr. Glaisher found at depths of 0, 3·2, 6·4, 12·8 and 25·6 feet, the difference between the mean of the hottest and coldest months to be 30°·5, 23°, 16°·1, 9°·31 and 3°·2 respectively. At 12·8 feet the mean temperature of January was 13° higher, and of July 15° lower than at the surface. It is surprising that this means of obtaining a moderate temperature in summer is not more extensively used:—a cage lowered by a windlass into a dry well to a depth of about 30 feet would give a constant temperature of about 50°, or about 20° below the mid-day summer heat in this country.

(44.) Below the stratum of invariable temperature, the heat increases at a rate which varies with the character of the soil: three artesian bores with depths from 1792 to 2400 feet gave 57·3 feet per degree, six Cornish copper and tin mines from 1380 to 2112 feet in granite gave 57·4 feet, and five others from 768 to 1530 feet in "Killas" gave 38·5 feet only, per degree.

Taking 58 feet per degree, and assuming 50° for the surface temperature, water would boil at $(212 - 50) \times 58 \div 5280 = 1{\cdot}78$ miles, and cast iron would melt at $(2010 - 50) \times 58 \div 5280 = 21{\cdot}5$ miles. The high temperature which thus arises is the great obstacle to deep mining: a coal-pit in Durham 1800 feet deep has a temperature of 80°, and a copper mine in Cornwall 2112 feet, has a temperature of 90°. At greater depths the heat would become intolerable to the men.

(45.) "*Sources of Heat and Cold.*"—The principal sources of

heat are: solar radiation (40); the internal heat of the globe (44); friction (46); compression (47); change of state from liquid to solid (13), and from vapour to liquid (17); chemical combinations (53), and combustion (57).

The principal sources of cold are: sky radiation (41); evaporation (193); dilatation of compressed air (51); and the use of ice and frigorific mixtures (53).

(46.) "*Heat developed by Friction.*"—The fact, that friction produces heat has been known from the earliest times, but the power required to generate a given amount of heat was not accurately ascertained until recently. By an agitator working in water and actuated by a falling weight, Mr. Joule found that to produce a "unit" of heat, required 772 foot-pounds, or 772 pounds falling 1 foot: this is termed the "mechanical equivalent" of heat. Conversely, one unit of heat should be capable of doing 772 foot-pounds of work in a steam-engine, &c., and as by (60) a pound of coal contains 13,000 units, it should yield $13000 \times 772 = 10{,}036{,}000$ foot-pounds. But the highest duty of the best Cornish pumping-engine does not exceed 107½ millions of foot-pounds per bushel (94 lbs.) of coal, or 1,143,620 foot-pounds per pound of coal, or about one-ninth of the theoretical amount.

(47.) "*Heat developed by Compression.*"—The law of Marriotte (29) is true only so long as the temperature remains constant as stated; but every change of pressure and its accompanying change of volume is simultaneously accompanied by a change of temperature, compression causing increase, and dilatation decrease.

With atmospheric air, the ratio of the specific heat (6) under constant pressure being to that with constant volume by Table 5 as 1·41 to 1, and absolute zero being − 458·4 by (27), the relations of pressure and temperature are given by the following rule:—

$$t = \left(\left(\frac{p}{P}\right)^{\frac{\cdot 41}{1\cdot 41}} \times (458\cdot 4 + T)\right) - 458\cdot 4;$$

$$\text{or} \quad t = \left(\left(\frac{p}{P}\right)^{\cdot 2908} \times (458\cdot 4 + T)\right) - 458\cdot 4;$$

TABLE 35.—Of the HEAT produced by COMPRESSION of AIR, and COLD by DILATATION, the Volume at Atmospheric Pressure being 1·0 at the Temperature of 60° Fahrenheit.

Pressure. Above a Vacuum. Atmospheres.	Pressure. Above a Vacuum. Inches of Mercury.	Pressure. Above a Vacuum. Pounds per Sq. Inch.	Pressure. Above the Atmosphere in Pounds per Sq. Inch.	Volume.	Temperature of the Air throughout the Process.	Total Increase or Decrease of Temperature.
					°	°
·1667	5	2·45	..	3·56	−152·6	−210·6
·3333	10	4·9	..	2·18	− 83·8	−141·8
·5	15	7·35	..	1·634	− 36·7	− 94·7
·6667	20	9·8	..	1·333	+ 2·4	− 57·6
·8333	25	12·25	..	1·137	33·0	− 27·0
1·0	30	14·7	0·00	1·0000	60	00·0
1·1	33	16·17	1·47	·9346	74·6	+ 14·6
1·25	37·5	18·37	3·67	·8536	94·8	34·8
1·5	45·0	22·05	7·35	·7501	124·9	64·9
1·75	52·5	25·81	11·11	·6724	151·6	91·6
2·0	60	29·4	14·7	·6117	175·8	115·8
2·5	75	36·7	22·0	·5221	218·3	158·3
3·0	90	44·1	29·4	·4588	255·1	195·1
3·5	105	51·4	36·7	·4113	287·8	227·8
4·0	120	58·8	44·1	·3741	317·4	257·4
5·0	150	73·5	58·8	·3194	369·4	309·4
6·0	180	88·2	73·5	·2806	414·5	354·5
7·0	210	102·9	88·2	·2516	454·5	394·5
8·0	240	117·6	102·9	·2288	490·6	430·6
9·0	270	132·3	117·6	·2105	523·7	463·4
10·0	300	147·0	132·3	·1953	554	494
15·0	450	220·5	205·8	·1465	681	621
20·0	600	294·0	279·3	·1195	781	721
25·0	750	367·5	352·8	·1020	864	804

APPROXIMATE TABLE.

(1)	(2)	(3)	(4)	(5)	(6)	(7)
1·0	30·0	14·7	0·00	1·0	60	00·
1·15	34·5	16·91	2·21	·9	84	24
1·37	41·1	20·14	5·44	·8	109	49
1·65	49·5	24·25	9·55	·7	141	81
2·06	61·8	30·28	15·58	·6	179	119
2·66	79·8	39·10	24·40	·5	229	169
3·63	108·9	53·36	38·66	·4	294	234
5·48	164·4	80·56	65·86	·3	392	332
9·65	289·5	141·85	127·15	·2	540	480
25·72	771·6	378·08	363·38	·1	869	809

in which P and T are the *total* pressure (above vacuum) and temperature in degrees Fahr. in one case, and p and t the pressure and temperature in another case. The pressure may be taken in inches of mercury or pounds per square inch, &c., but of course must be the same in both cases.

Thus say we have a volume of air = 1·0 at 60°, with the barometric pressure = 30 inches of mercury, and we increase the pressure to 7 atmospheres or $30 \times 7 = 210$ inches above a vacuum; then $P = 30$, $p = 210$ and $T = 60$. To find the ·2908 power of $\frac{p}{P}$ we must make use of logarithms; then $210 \div 30 = 7$, the logarithm of which is ·845098 and $·845098 \times ·2908 = ·245754$, the natural number of which is 1·761. Having thus obtained $7^{·2908} = 1·761$ we have no difficulty in applying the rule, and we obtain $1·761 \times (458·4 + 60) - 458·4 = 454°·5$, the temperature sought, which shows that compression has produced an increase of temperature of $454·5 - 60 = 394°·5$. The volume may now be found by the rule in (27) which becomes in our case $1 \times \frac{30}{210} \times \frac{458·4 + 454·5}{458·4 + 60} = ·2516$, showing that the volume is reduced to one-fourth nearly. The cols. 5, 6, and 7 in Table 35 have been calculated in this way. The rules will not give the temperature and pressure due to compression into any given volume direct, because the unknown temperature and pressure are involved in the question, but by drawing a diagram we may obtain both with approximate correctness, and we have thus obtained the numbers in the second part of Table 35. Thus, by suddenly compressing air to one-tenth of its volume we should increase its pressure to 10 atmospheres by the law of Marriotte, but the compression causes an increase in the temperature of 809°, which increases the pressure from 10 atmospheres to 25·72 atmospheres as shown by col. 1 in the table.

(48.) The heat developed will vary slightly with the ratios of the specific heats of the particular gas operated on, but may be found with the same rule by substituting for the fraction $\frac{·41}{1·41}$, the ratios given by col. 3 of Table 5, and where great accuracy is desired substituting the true zero from col. 6 of Table 26,

or cols. 2 and 4 of Table 27 for 458·4 given by the rule. Thus, for carbonic acid gas with a density of 4·7318 the rule becomes

$$t = \left(\frac{p}{P}\right)^{\frac{\cdot 264}{1\cdot 264}} \times (434\cdot 3 + T) - 434\cdot 3, \text{ \&c.}$$

(49.) The temperatures given by Table 35 will seldom be fully realized in practice except under particular conditions, because the heat generated although high in temperature is small in quantity, owing to the low specific heat and weight of air. Thus a cubic foot of air at 62° weighs by Table 24 only ·0761 lb.; compressed to one-tenth of its volume it would be heated 809°, and the specific heat being ·238, we have ·0761 × 809 × ·238 = 14·65 units of heat developed: then if the air-pump, &c., weighed 100 lbs., and the specific heat of cast iron being ·13 nearly, the 14·65 units would be absorbed by it, and its own temperature would be increased only 14·65 ÷ (·13 × 100) = 1°·13. But where the compression is continuously renewed as with an air-pump regularly working, the temperatures due thereto would be approximately attained.

(50.) The heat thus evolved by the compression of air is found very troublesome in some cases; such as air-pumps for diving bells, pneumatic piles of bridges, &c., being destructive to the lubricating oil. The pumps may be kept cool by surrounding them with cold water continually changed by a current passing through the cistern, and we can easily calculate the quantity of water necessary. Say we have double-acting pumps with two barrels 12 inches diameter 18 inches stroke, thirty-five revolutions per minute, discharging 165 cubic feet of air (taken at the atmospheric pressure) against 100 feet head of water, or 88·5 say 90 inches of mercury by Table 38. The *total* pressure is thus increased from 30 to 30 + 90 = 120 inches of mercury, or from 1 atmosphere to 120 ÷ 30 = 4 atmospheres, and by col. 7 of Table 35 the heat developed is 257°. The weight of the air is ·0761 × 165 = 12·6 lbs., and its specific heat being that with constant pressure (5) or ·238, the heat developed will be 12·6 × 257 × ·238 = 770 units, and admitting that the water used for cooling may be heated 30°, we require 770 ÷ 30 = 25·7 lbs., or 2·57 gallons per minute.

(51.) "*Freezing by Compressed Air.*"—If compressed air be deprived by cold water or otherwise of the heat developed by compression, and then suffered to return to its normal volume by the relief of the pressure, a very low temperature is obtained, which may be used for freezing water and other purposes.

The amount of heat to be taken from a pound of water at 60° to reduce it to ice at 32° is 170 units, namely,

	Units.
One pound of water at 60° to water at 32°	28
" " 32° " ice " 32° (latent)	142
	170

Assuming the pressure of the compressed air at 2 atmospheres total, we can calculate the quantity of air necessary to freeze a pound of water, and the mechanical power required to do the work. One pound of air at 1 atmosphere and at 60° compressed to 2 atmospheres, is heated 116° by col. 7 of Table 35, and the specific heat of air where expansion is permitted being $\cdot 238$ we have $\cdot 238 \times 116 = 27\cdot 6$ units per pound of air, and to freeze a pound of water from 60° requires $170 \div 27\cdot 6 = 6\cdot 16$ lbs. of air, or $6\cdot 16 \div \cdot 0761 = 81$ cubic feet of air at 1 atmosphere and at 60°.

We now have to find the power required to compress this air. Imagine an air-pump 1 foot square and 1 foot stroke, thus holding 1 cubic foot, and let it discharge air compressed to 2 atmospheres, and heated 116° by the compression, into a reservoir where it is cooled down by cold water to 60° again. Let that compressed air be caused to pass through a refrigerator where it is allowed to return to its normal pressure and in so doing to absorb from the water to be frozen, the heat which it gave out when compressed, and let the pressure in the reservoir be maintained uniformly at 2 atmospheres.

(52.) Now the pressure on the piston of the air-pump would be nothing at the commencement of the stroke, rising to 2 atmospheres above a vacuum when the volume was reduced to $\cdot 6117$ cubic foot by col. 5 of Table 35, or when the piston had travelled $1 - \cdot 6117 = \cdot 3883$ foot. To find the power expended

in this part of the operation we must take the travel of the piston given by col. 5 and the *mean* pressure from col. 4: thus, the piston travels $1 - \cdot 9346 = \cdot 0654$ foot with a mean pressure of $(0 + 1 \cdot 47) \div 2 = \cdot 735$ lb. per square inch requiring $\cdot 0654 \times \cdot 735 = \cdot 048$ foot-pound; the next travel is $\cdot 9346 - \cdot 8536 = \cdot 081$ foot, against $(1 \cdot 47 + 3 \cdot 67) \div 2 = 2 \cdot 57$ lbs. requiring $\cdot 081 \times 2 \cdot 57 = \cdot 208$ foot-pound. We thus obtain the following numbers:—

	Foot-pounds.
$(1 \cdot 0000 - \cdot 9346) \times (0 \cdot 00 + 1 \cdot 47) \div 2 =$	$\cdot 048$
$(\cdot 9346 - \cdot 8536) \times (1 \cdot 47 + 3 \cdot 67) \div 2 =$	$\cdot 208$
$(\cdot 8536 - \cdot 7501) \times (3 \cdot 67 + 7 \cdot 35) \div 2 =$	$\cdot 570$
$(\cdot 7501 - \cdot 6724) \times (7 \cdot 35 + 11 \cdot 11) \div 2 =$	$\cdot 717$
$(\cdot 6724 - \cdot 6117) \times (11 \cdot 11 + 14 \cdot 7) \div 2 =$	$\cdot 783$
Total =	$2 \cdot 326$

We thus find the power during the first part of the stroke to be $2 \cdot 326$ foot-pounds per square inch of piston, or $2 \cdot 326 \times 144 = 335$ foot-pounds on the whole area. The piston then completes the stroke or $\cdot 6117$ foot against a uniform pressure of $14 \cdot 7 \times 144 = 2116 \cdot 8$ lbs. per square foot, and requires $2116 \cdot 8 \times \cdot 6117 = 1295$ foot-pounds more, making a total of $335 + 1295 = 1630$ foot-pounds to compress and deliver a cubic foot of air taken at atmospheric pressure, &c., and the 81 cubic feet which we found necessary (51) to freeze a pound of water, required $1630 \times 81 = 132030$ foot-pounds.

By Joule's experiments (46) the mechanical equivalent of a unit of heat is 772 foot-pounds; hence the 170 units necessary to be taken from a pound of water in order to freeze it required $170 \times 772 = 131240$ foot-pounds, which is very nearly the amount we have calculated.

The indicated horse-power being equal to 33,000 foot-pounds per minute, we find that one horse-power would produce $33000 \times 60 \div 132030 = 15$ lbs. of ice per hour. But we have here allowed nothing for the friction of the air-pumps; perhaps 10 lbs. of ice would be about the amount in practice, and allowing as in (74), $5 \cdot 75$ lbs. of coal per indicated horse-

power, we have $10 \div 5 \cdot 75 = 1 \cdot 75$ lbs. of ice per pound of coal. By placing another cylinder between the refrigerator and the open air, so as to utilize the pressure left in the air after cooling, the power might be considerably reduced, but the complication of machinery and consequent loss by friction would neutralize to a great extent the advantage thus gained.

(53.) "*Frigorific Mixtures.*"—The mixture of many salts with water, snow, and acids is productive of cold, and by this means a very low temperature may be obtained at all seasons and in all climates. Table 36 gives a few of the best mixtures for this purpose: most of them may be recovered after use by evaporation. The best practical mixture is that of common salt with snow or pounded ice, by which an intense cold, 32° below the freezing point of water may be easily obtained.

"*Heating Mixtures.*"—The development of heat is a frequent result of the mixture of liquids with one another or with solids: thus equal volumes of sulphuric acid and water, both at 57°, give 212° as the result of mixture. When quicklime is slaked with water, a very high temperature is generated, varying however very much with the quantity of water used. Fresh burnt chalk-lime was found to absorb about 2½ times its own weight of water, and if much more than that is used, the heat developed

TABLE 36.—OF FRIGORIFIC MIXTURES for the ARTIFICIAL PRODUCTION of COLD, from the Experiments of Mr. WALKER.

	Proportional Parts, by Weight, in the Mixture.											
Water	16	1	..	..	..	..	..	..	..	..	..	..
Sal Ammoniac	5	..	..	..	..	..	..	..	..	1	5	..
Nitre	5	..	..	..	..	..	..	..	..	..	5	..
Common Salt	..	..	..	..	1	..	..	..	1	2	10	5
Nitrate of Ammonia	..	1	5	6	..	..	..	..	..	..	..	5
Sulphate of Soda	8	..	6	..	..	..	..	..	..	..	..	..
Carbonate of Soda	..	1	..	..	..	..	..	..	..	..	..	..
Phosphate of Soda	..	..	..	9	..	..	..	..	..	..	..	..
Potash	..	..	..	..	..	..	..	4	..	..	..	..
Muriate of Lime	..	..	..	..	..	..	5	..	..	..	..	..
Snow or pounded Ice	..	..	..	..	1	8	4	3	2	5	24	12
Diluted Nitric Acid	..	..	4	4	..	..	..	..	..	..	..	..
Diluted Sulphuric Acid	..	..	..	..	..	10	..	..	..	..	..	.
Temperature of Ingredients	+50	+50	+50	+50	+32	−68	+32	+32	..	..	..	..
Temperature of the Mixture	+ 4	− 7	−14	−21	0	−91	−40	−51	−5	−12	−18	−25
Cold produced	46°	57	64	71	32	23	72	83	..	..	..	..

NOTE.—To obtain these results the temperature of the ingredients must be reduced previously to the given temperature by some of the other mixtures. The four last compounds yield the temperatures given, whatever may be the previous temperatures of the ingredients.

is very small: thus with water 6 to lime 1, the water was heated from 52° to 60°, or 8° only, but with equal weights the temperature became 210°. With less water the high temperature of 476° was obtained, varying however in different parts of the mass: two other experiments gave 412° and 384° respectively.

(54.) "*Density and Weight of Bodies.*"—The specific gravity or weight of bodies is frequently required throughout this work, and it will be convenient to give it in a collected form. This is done for solids and liquids in Table 37, the basis of

TABLE 37.—Of the SPECIFIC GRAVITY and WEIGHT of MATERIALS, Water at 62° being 1·000.

	Specific Gravity.	Weight of a Cubic Foot in Pounds.	Weight of a Cubic Inch in Pounds.	No. of Cubic Feet in One Ton.
Mercury	13·596	847·3	·4903	2·644
Lead	11·352	707·5	·4094	3·166
Copper, sheet	8·785	547·5	·3168	4·091
Gun Metal, cast	8·670	540·3	·3127	4·145
Copper, cast	8·607	536·4	·3104	4·176
Brass, cast	8·393	523·1	·3027	4·282
Wrought Iron	7·788	485·3	·2809	4·615
Tin, cast	7·291	454·4	·2630	4·930
Zinc, sheet	7·190	448·1	·2593	4·999
Cast Iron, British, mean	7·087	441·6	·2556	5·07
Zinc, cast	6·861	427·6	·2474	5·24
Slate	2·835	176·7	·1022	12·68
Glass	2·760	172·0	·0995	13·02
Granite, Cornish	2·662	165·9	·0960	13·50
Sandstone, Yorkshire	2·506	156·2	·0904	14·34
Brick, London Stock	1·841	114·7	·0664	19·52
Sand, River	1·546	96·35	·0558	23·25
Coal, British, mean	1·313	81·83	·0474	27·37
Water, distilled	1·000	62·321	·03606	35·95
Ice, at 32°	·93	57·96	·03354	38·65
Alcohol	·813	50·67	·02932	44·21
Oil, Olive	·9153	57·04	·03301	39·27
Oak, seasoned	·777	48·42	·02802	46·26
Elm, "	·588	36·65	·0212	61·15
Mahogany, Honduras, seasoned	·560	34·9	·0202	64·18
Pine, Yellow, seasoned	·483	30·1	·01742	74·41
Coke, Gas, in measure	·353	22·0	·01273	101·8
Cork	·24	14·96	·00866	149·7

which is the weight of distilled water at 62°, when by Act of Parliament a gallon containing 277·274 cubic inches weighs 10 lbs.; hence a cubic foot at that same temperature weighs $1728 \times 10 \div 277 \cdot 274 = 62 \cdot 321$ lbs. The weight of water at other temperatures is given by Table 21, and Table 38 gives the pressure of columns of water in pounds per square inch, and in inches of mercury.

TABLE 38.—Of EQUIVALENT PRESSURES in POUNDS per SQUARE INCH, FEET of WATER, and INCHES of MERCURY, at a Temperature of 62° Fahr.

Pounds per Square Inch.	Feet of Water.	Inches of Mercury.	Pounds per Square Inch.	Feet of Water.	Inches of Mercury.
1·	2·311	2·046	2·5962	6·	5·31198
2·	4·622	4·092	3·0289	7·	6·19731
3·	6·933	6·138	3·4616	8·	7·08264
4·	9·244	8·184	3·8942	9·	7·96797
5·	11·555	10·230	·48875	1·12952	1·
6·	13·866	12·276	·97750	2·25904	2·
7·	16·177	14·322	1·46625	3·38856	3·
8·	18·488	16·368	1·95500	4·51808	4·
9·	20·800	18·414	2·44375	5·64760	5·
·4327	1·	·88533	2·93250	6·77712	6·
·8654	2·	1·77066	3·42125	7·90664	7·
1·2981	3·	2·65599	3·91000	9·03616	8·
1·7308	4·	3·54132	4·39875	10·16568	9·
2·1635	5·	4·42665			

EXAMPLE.—Required the Pressure per Square Inch, and Equivalent Column of Mercury for a Head of 247 feet of Water.

Feet of Water.		Pounds per Square Inch.		Inches of Mercury.
200	=	86·54	or	177·066
40	=	17·308	„	35·413
7	=	3·029	„	6·197
247	=	106·877	„	218·676

(55.) The density and weight of gases are complicated by temperature and pressure. Table 39 gives the density, &c., at 62° under a pressure of 29·92 inches of mercury in the barometer. The results for vapours in cols. 2, 3, 4, 5 are somewhat fictitious, because the relations of vapours as to pressure and temperature are fixed and unalterable. With a pressure of

29·92 inches of mercury, as in Table 39, the vapour of water can have no other temperature than 212°; and ether 100°, as in Table 10. At 62° water has an elastic force of ·556 inch of mercury only, instead of 29·92 inches; but we can calculate the density by col. 1, which gives the ratios to air at *all* temperatures and pressures.

Table 68 gives in col. 7 the true weight of a cubic foot of vapour at its own normal and proper pressure, given by col. 4, both being governed inflexibly by the temperatures in col. 1 in the same table (79). The numbers in Table 39 are given to facilitate calculation, the correctness of which will not be affected by the fact that the data are fictitious. Thus col. 4 gives the weight of vapour of water at 62° and 29·92 inches pressure, at ·04745 lb. per cubic foot; at its true normal pressure of ·556 inch, the weight would be $\cdot04745 \times \cdot556 \div 29\cdot92 = \cdot000881$ lb., as per col. 7 of Table 68. The volume and weight of dry air at different temperatures is given by Table 24.

TABLE 39.—Of the DENSITY of GASES and VAPOURS, Air at the same Temperature and Pressure being 1·0; also the Weight of a Cubic Foot at 62°, under an Atmospheric Pressure of 29·92 inches of Mercury.

	Density, Air at the same Temp. and Pressure being 1. (Regnault.)	Specific Gravity or Density, Water at 62° being 1·0.		Weight of a Cubic Foot in Pounds.	Cubic Feet at 62° in 1 lb.
Air (atmospheric)	1·00000	·001221 or	$\frac{1}{819}$	·07610	13·14
Hydrogen Gas	·06926	·0000846 „	$\frac{1}{11820}$	·00527	189·70
Oxygen Gas	1·10563	·001350 „	$\frac{1}{741}$	·08414	11·88
Nitrogen Gas	·97137	·001185 „	$\frac{1}{844}$	·07383	13·54
Carbonic Acid Gas	1·52901	·001870 „	$\frac{1}{535}$	·11636	8·59
Carbonic Oxide Gas ..	·9674	·00118 „	$\frac{1}{847}$	·07364	13·60
Vapour of Water	·6235	·0007613 „	$\frac{1}{1313}$	·04745	21·07
„ Alcohol	1·589	·00194 „	$\frac{1}{515}$	·12092	8·27
„ Sulphuric Ether	2·586	·00316 „	$\frac{1}{316}$	·19680	5·08
„ Oil of Turpentine	4·760	·00581 „	$\frac{1}{172}$	·36224	2·76
„ Mercury	6·976	·00850 „	$\frac{1}{118}$	·52987	1·88
	(1)	(2)	(3)	(4)	(5)

NOTE.—The densities of the *vapours* in column (2), &c., are reduced from their real pressure and density by calculation.

(56.) "*Atomic Weights of Bodies.*"—All bodies which combine chemically tend to do so in fixed definite proportions, which are termed "atomic weights," or "chemical equivalents," and are given for a few elementary bodies in Table 40. Thus 1 atom of oxygen, or 100 lbs., combining with 2 atoms of nitrogen, or 350 lbs., forms 100 + 350 = 450 lbs. of atmospheric air. Again, 5 atoms, or 500 lbs., of oxygen, combining with 1 atom, or 175 lbs., of nitrogen, forms 500 + 175 = 675 lbs. of nitric acid. Other illustrations of the application of the table are given in (3) (57), &c.

TABLE 40.—Of the ATOMIC WEIGHTS of ELEMENTARY BODIES, Oxygen being = 100.

Element.	Combining Weights.	Element.	Combining Weights.	Element.	Combining Weights.
Aluminium	171·16	Gold ..	1243·01	Oxygen ..	100·00
Antimony	1612·90	Hydrogen	12·50	Phosphorus	392·31
Bismuth ..	1330·37	Iron	339·20	Platina ..	1233·50
Calcium ..	256·02	Lead ..	1294·50	Silver ..	1351·61
Carbon ..	75·00	Mercury ..	1265·82	Sulphur ..	201·16
Chlorine ..	442·65	Nickel ..	369·67	Tin	735·29
Copper ..	395·69	Nitrogen ..	175·00	Zinc	403·23

CHAPTER II.

ON COMBUSTION.

(57.) "*Theory of Combustion.*"—Combustion consists in the combination of bodies, and principally of carbon and hydrogen with oxygen, the carbon combining with oxygen derived from the air forming carbonic acid, and hydrogen similarly combining forming water. One "atom," or 75 lbs., of carbon (Table 40), combining with 2 atoms, or 200 lbs., of oxygen, forms 275 lbs. of carbonic acid. One atom, or 12·5 lbs., of hydrogen, combining with 1 atom, or 100 lbs., of oxygen, forms 112·5 lbs. of water. By the experiments of Dulong, the heat evolved by

these combinations is 12,906 units per pound of carbon, and 62,535 units per pound of hydrogen.

The combustibles used in the arts are principally composed of carbon and hydrogen, as shown by Table 41, and from their known composition we can easily calculate the heat developed by them. Thus, oil of turpentine, by the table, consists of ·884 carbon and ·116 hydrogen, and 1 lb. of it will yield—

Carbon ..	$\cdot 884 \times 12906 =$	11409	units of heat
Hydrogen ..	$\cdot 116 \times 62535 =$	7254	"
		18663	"

TABLE 41.—Of the CHEMICAL COMPOSITION of COMBUSTIBLES, according to PÉCLET, &c.

Elements.	Carbon.	Hydrogen.	Oxygen.	Nitrogen and Sulphur.	Water.	Ashes.	Total.
Coal, mean 97 kinds	·804	·0519	·0787	·0246	..	·0408	1·000
Coke	·850	..	..	..	..	·150	"
Wood, perfectly dry	·510	·053	·417	..	..	·020	"
" ordinary state	·408	·042	·334	..	·200	·016	"
" charcoal ..	·930	..	..	..	..	·070	"
Peat, perfectly dry	·580	·060	·310	..	..	·050	"
" ordinary state	·464	·048	·248	..	·200	·040	"
Oil of Turpentine ..	·884	·116	..	..	..	..	"
Alcohol	·5198	·1370	·3432	..	..	..	"
Olive Oil..	·7721	·1336	·0943	..	..	..	"
Sperm Oil	·789	·1097	·1013	..	..	..	"
Beeswax	·816	·139	·045	..	..	..	"
Spermaceti	·816	·128	·056	..	..	..	"
Tallow	·790	·117	·093	..	..	..	"
Paraffine Oil	·8522	·1478	..	..	..	..	"
Resin	·7927	·1015	·1058	..	..	..	"
Sulphuric Ether ..	·6531	·1333	·2136	..	..	..	"

(58.) The presence of oxygen in a combustible containing hydrogen has the effect of reducing its heating power, for when a combustible contains 8 lbs. of oxygen to 1 lb. of hydrogen, being the ratio in which they combine to form water, they do so combine, but give out no useful heat, whereas if hydrogen alone is present it yields the full amount of heat due to it. When oxygen is present, but in too small a proportion to combine

with the *whole* of the hydrogen, it combines with one-eighth of its weight, and leaves the rest as an *excess of hydrogen*, which yields its due proportion of heat as before.

(59.) Alcohol will serve to illustrate the effect of oxygen in a combustible, it being composed, as per Table 41, of ·5198 carbon, ·137 hydrogen, and ·3432 oxygen. We have seen in (57) that oxygen requires one-eighth part of its weight of hydrogen to form water; the oxygen in alcohol will require $\cdot 3432 \div 8 = \cdot 0429$ hydrogen; whereas the combustible contains ·137 of hydrogen: there remains, therefore, $\cdot 137 - \cdot 0429 = \cdot 0941$ hydrogen *in excess* to develop its heat, and we have—

Carbon	$\cdot 5198 \times 12906 =$	6708 units
Hydrogen in excess ..	$\cdot 0941 \times 62535 =$	5885 „
		12593

Similarly for Olive Oil, composed of ·7721 carbon, ·1336 hydrogen, and ·0943 oxygen, we have—

Carbon	$\cdot 7721 \times 12906 =$	9965 units
Hydrogen $\left(\cdot 1336 - \frac{\cdot 0943}{8}\right) =$	$\cdot 1218 \times 62535 =$	7517 „
		17482 „

Tallow, composed of ·79 carbon, ·117 hydrogen, and ·093 oxygen, will give—

Carbon	$\cdot 79 \times 12906 =$	10195 units
Hydrogen $\left(\cdot 117 - \frac{\cdot 093}{8}\right) =$	$\cdot 1055 \times 62535 =$	6597 „
		16792 „

Paraffine Oil, or Petroleum, composed of carbon ·8522 and hydrogen ·1478, will give—

Carbon	$\cdot 8522 \times 12906 =$	10998 units
Hydrogen	$\cdot 1478 \times 62535 =$	9242 „
		20240 „

Sulphuric Ether, composed of carbon ·6531, hydrogen ·1333, and oxygen ·2136, will give—

$$\text{Carbon} \quad \ldots \quad \cdot 6531 \times 12906 = 8429 \text{ units}$$

$$\text{Hydrogen} \left(\cdot 1333 - \frac{\cdot 2136}{8} \right) = \cdot 1066 \times 62535 = 6666 \quad ,,$$

$$15095 \quad ,,$$

Beeswax, composed of carbon ·816, hydrogen ·139, and oxygen ·045, will give—

$$\text{Carbon} \quad \ldots \quad \cdot 816 \times 12906 = 10531 \text{ units}$$

$$\text{Hydrogen} \left(\cdot 139 - \frac{\cdot 045}{8} \right) = \cdot 1334 \times 62535 = 8342 \quad ,,$$

$$18873 \quad ,,$$

(60.) From the Government experiments on Coal, by Playfair and De la Bèche, we find, as a mean of 97 different kinds of English, Welsh, and Scotch coals, the composition to be as per Table 41, and from this we have—

$$\text{Carbon} \quad \ldots \quad \cdot 804 \times 12906 = 10376 \text{ units}$$

$$\text{Hydrogen} \left(\cdot 0519 - \frac{\cdot 0787}{8} \right) = \cdot 04206 \times 62535 = 2630 \quad ,,$$

$$13006 \quad ,,$$

Coke, containing ·85 carbon, without any hydrogen or oxygen, will yield ·85 × 12906 = 10970 units of heat.

(61.) Wood, perfectly dry, contains not only ·51 carbon, but also ·47 hydrogen and oxygen, but as these are in the proportion proper for forming water, they combine without yielding any useful heat, and we have ·51 × 12906 = 6582 units per pound of dry wood.

Wood in its ordinary state of dryness contains 20 per cent. of water, its carbon is thereby reduced to ·51 × ·8 = ·408, and its calorific power to ·408 × 12906 = 5265 units per pound.

Charcoal from wood, containing ·93 carbon, will give ·93 × 12906 = 12000 units per pound.

(62.) Peat, artificially dried, contains ·58 carbon, ·06 hydrogen, and ·31 oxygen. The oxygen in the fuel will combine with ·31 ÷ 8 = ·04 hydrogen, leaving ·06 − ·04 = ·02 hydrogen in *excess*, to develop its share of heat; and we have—

Carbon	·58 × 12906 =	7485 units
Hydrogen in excess ..	·02 × 62535 =	1251 „
		8736 „

Peat in its natural state of dryness contains ·464 carbon, ·048 hydrogen, and ·248 oxygen. The oxygen will combine with ·248 ÷ 8 = ·031 hydrogen, leaving ·048 − ·031 = ·017 hydrogen in excess, and we have—

Carbon	·464 × 12906 =	5988 units
Hydrogen in excess ..	·017 × 62535 =	1163 „
		7151 „

Charcoal of peat contains ·818 carbon, and yields ·818 × 12906 = 10557 units per pound.

The heating power of combustibles, as found by the preceding calculations, is the maximum effect they are capable of producing. When we come to apply it to practice we shall see (103) that there are sources of unavoidable loss which reduce their useful effect considerably.

(63.) "*Effect of Water in a Combustible.*"—When a combustible contains water, with which it is more or less saturated, the effect is twofold; say we take wood in its ordinary state, containing 20 per cent. of water; the really combustible matter is reduced to 80 per cent. of the amount contained in the same weight of wood perfectly dry, and of necessity the calorific power is reduced in the same proportion. The second effect is, that part of the heat in the residue is consumed uselessly in evaporating the water. Wood perfectly dry gives by Table 42, 6480 units,

which for wood in the ordinary state containing 20 per cent. of water, is reduced to $6480 \times \cdot 8 = 5184$ units, but the $\cdot 2$ water, say at 62°, will require for its evaporation $(1178 - 62) \times \cdot 2 = 223$ units, so that the useful heat is reduced to $5184 - 223 = 4961$ units: this correction may be neglected in many cases, the amount being small in proportion to the heat given out by the fuel.

The experiments of Rumford have shown that the heating power of wood varies only with the state of dryness; that is to say, all the different kinds of wood in the same state of dryness yield sensibly the same amount of heat.

(64.) "*Experimental Power of Combustibles.*"—The heating power of combustibles may also be determined by direct experiment with "calorimeters" specially designed for the purpose, such as Rumford's and others. Rumford's apparatus consisted of a shallow vessel of copper filled with water; the fuel to be experimented on is burnt beneath it, and the products of combustion being collected by a hood or inverted funnel, are caused to pass by a worm circulating through the mass of the water, and the calorific power is estimated by the increase in temperature of the water and the vessel. To avoid loss of heat by the apparatus during the experiment, the *mean* temperature is arranged to be the same as the temperature of the air; thus if the air was at 60° and the range of temperature 20°, the water would be taken at 50° to begin with, and would be 70° at the end of the experiment. With such an apparatus we should not obtain the total heating power in the fuel; some of the heat would be lost by direct radiation from the burning fuel, and some would pass off with the air issuing from the apparatus after passing through it, which would always have a temperature higher than the water it was intended to heat.

(65.) Say we had an apparatus like Rumford's, made of sheet copper, weighing 10 lbs. and holding 60 lbs. of water at 50°, which became heated to 70° by $\frac{1}{4}$ of a pound or 4 ounces of dry wood. The specific heat of copper (Table 1) being $\cdot 095$ by Regnault's experiments, the amount of heat absorbed by the vessel heated 20° will be $10 \times 20 \times \cdot 095 = 19$ units, and the water absorbs $60 \times 20 = 1200$ units. The $\frac{1}{4}$ lb. of fuel has

thus given out 19 + 1200 = 1219 units, which is equal to 1219 ÷ ·25 = 4876 units per pound. Table 42 gives the calorific power of combustibles by Rumford and others.

TABLE 42.—Of the CALORIFIC POWER of COMBUSTIBLES, by Philosophical Experiment and Theory.

	Units of Heat per lb. of Fuel, by Experiment.	Authorities.	Units of Heat per lb. of Fuel by Theory.
Hydrogen, burning to Water	62535	Dulong.	
" "	62032	Favre & Silbermann.	
" "	42120	Laplace.	
" "	39807	Clement.	
" "	42552	Despretz.	
Carbon, burning to Carbonic Acid	12906	Dulong.	
" from wood, burning to Carbonic Acid	14544	Favre & Silbermann.	
Carbon, burning to Carbonic Acid	14040	Despretz.	
" " Carbonic Oxide	2495	Dulong.	
" " "	4453	Favre & Silbermann.	
Carbonic Oxide, burning to Carbonic Acid	4478	Dulong.	
" " "	4325	Favre & Silbermann.	
1 lb. Carbon in the form of Carbonic Oxide, burning to Carbonic Acid	10411	Dulong.	
Wood, perfectly dried by artificial heat	6480	Rumford	6582
Wood, in ordinary state of dryness	5040	"	5265
Olive Oil	17752	Dulong	17482
"	16279	Rumford.	
"	20153	Lavoisier.	
Colza Oil	16753	Rumford.	
Alcohol	12339	Dulong	12593
"	11151	Rumford.	
Sulphuric Ether	16974	Dulong	15095
" " density ·728 at 68°	14454	Rumford.	
Tallow	15550	"	16792
"	12935	Laplace.	
Oil of Turpentine	19505	Dulong	18663
Naphtha, density ·827	13208	Rumford	
Sulphur	4682	Dulong.	
"	4032	Favre & Silbermann.	
Phosphorus	13500	Laplace.	
Beeswax, white	18900	"	18873
" "	17422	Rumford.	

(66.) "*Practical Experiments on Fuel.*"—The most valuable experiments are those that have been made on the large scale on steam-engine boilers in practice; we then obtain the heating power of fuel in a form directly applicable to cases analogous to the experimental ones.

"*Power of Wood.*"—The best experiments on the heating power of wood, on the large scale, are given by Péclet. In one case with a hot-water boiler in a public bath, 15,800 lbs. of water were heated 153° by 440 lbs. of wood in the ordinary state of dryness, containing 20 per cent. of water; this gives $15800 \times 153 \div 440 = 5494$ units per pound of wood. This is an exceptionally good result, arising no doubt from the fact that the apparatus was so well arranged that the smoke left it at the temperature of the atmosphere, or nearly so, the whole of the heat given out by the wood being thus utilized.

(67.) In another experiment on a steam-boiler, 3·24 lbs. of water were evaporated per pound of wood: the air passed into the chimney at 480°, and retained half of its oxygen, or 10 per cent. unconsumed (76). The temperature of the feed-water is not given, but assuming that it was 100°, the calorific power of the wood is $3 \cdot 24 \times (1178 - 100) = 3493$ units per pound of fuel.

This is a low result, but it is what might be expected under the circumstances. The wood used contained 25 per cent. of water, therefore a pound of such wood contained only ·75 lb. of real combustible; the water would require $\cdot 25 \times (1178 - 62°) = 279$ units to evaporate it, and there would be a further loss of heat by the air in the chimney which departs highly heated. We shall see in (77) that *dry* wood requires 161 cubic feet of air when the oxygen is only half consumed, as in our case; therefore wood containing 25 per cent. of water requires $161 \times \cdot 75 = 120$ cubic feet; or $120 \times \cdot 0761 = 9 \cdot 1$ lbs. of air, which, heated 420°, or say from 60° to 480°, the temperature of the air in the chimney will carry off $9 \cdot 1 \times 420 \times \cdot 238 = 910$ units. Adding these together, we obtain $3493 + 279 + 910 = 4682$ units per pound of damp wood containing 25 per cent. of water, which is equal to $4682 \div \cdot 75 = 6242$ units per pound of wood perfectly dry. Rumford's experiments in Table 42 give 6480 units.

The circumstances under which this experiment was made are analogous to most practical cases, and the heating power of wood in its ordinary state of dryness may be taken at about 3500 units per pound of fuel.

(68.) "*Power of Coals.*"—Experiments were made upon the *weekly* consumption of coals by two Elephant or French boilers, composed of a body of large diameter, connected by necks with three smaller ones about 20 inches diameter: the furnaces were Juckes' self-acting. The experiments were made with two kinds of coal, one a kind of Welsh coal, and the other, small coal, or screenings from Yorkshire coals; each kind was experimented on for six days, from Monday to Saturday, and the result includes, therefore, loss by radiation during the night, &c., &c., and in getting up steam each morning. The water evaporated was *measured* in a vessel from which the feed-pump was supplied.

(69.) With the Welsh coals 193,876 lbs. of water were evaporated by 28,160 lbs. of coals, or $193876 \div 28160 = 6 \cdot 88$ lbs. of water per pound of coal. The water had a temperature of about 80°, and hence we have $(1178 - 80) \times 193876 \div 28160 = 7560$ units per pound of Welsh coals.

(70.) With the Yorkshire small coals, 167,526 lbs. of water were evaporated by 29,802 lbs. of coal, or $167526 \div 29802 = 5 \cdot 621$ lbs. of water per pound of coal, or $(1178 - 80) \times 167526 \div 29802 = 6172$ units of heat per pound of Yorkshire small coal.

Both these results are low, as might be expected, but they apply without correction to analogous cases, which are very numerous. The engine *worked* about twelve hours per day.

(71.) There is a source of error in such experiments, frequently overlooked, arising from *priming*, in which water passes over with the steam, and is not evaporated to steam at all. In the two following experiments this was avoided by taking off the man-hole cover and allowing the steam to escape into the atmosphere; the water was measured in by hand with a two-gallon spirit measure.

An experiment was made with Cater's patent tubular boiler, in which the fire is placed beneath the body, passes along the

bottom, returning to the front by a set of 4-inch tubes, and again to the back by another set of 3-inch tubes, thus traversing the length of the boiler thrice; the temperature of the air as it left the boiler was 320°. In six hours, 628 lbs. of Duffryn Welsh coals evaporated 5140 lbs. of water at 52° to steam at 212°, and we have therefore $(1178 - 52) \times 5140 \div 628 = 9220$ units per pound of coal.

(72.) This boiler was full 20 horse-power, but could not be worked up to more than 10-horse in the experiment, because it primed or boiled over at the man-hole, when the fire was kept up to its proper intensity. This is what might have been expected; by Table 71, each pound of water evaporated, formed 1640 cubic feet of steam, whereas with say 45 lbs. pressure, only 439 cubic feet would have been formed, and the tendency to boil over would be proportional to the volume of steam to be extricated from the water in a given time. To produce the same amount of priming with 45 lbs. steam, this boiler would require to be worked up to $10 \times 1640 \div 439 = 37$ horse-power. The result was, that the fire-grate could not be kept covered with fuel, and the economic result was not as good as it might have been if the boiler could have been worked up to its full power.

(73.) A similar experiment was made with a Cornish boiler, the fire being inside as usual. In six hours, 361 lbs. of Duffryn coals evaporated 2600 lbs. of water at 58° to steam at 212°, and we have therefore $(1178 - 58) \times 2600 \div 361 = 8066$ units per pound of coal.

(74.) The best and most extensive series of experiments on coal, are those made by H.M. Commissioners with a Cornish boiler of about 7 horse-power, the general results of which together with a *résumé* of the preceding experiments are given by Table 43. The mean heat utilized per pound of coal varied from 8742 units with Welsh coal, to 7322 units with Derbyshire, the mean of which extremes is 8033 units, and this is very nearly the effect of Newcastle coal which is given by the table at 8085 units.

The third column of Table 43 may be taken to represent pounds of fuel per nominal horse-power per hour (118); thus to raise a cubic foot of water from 60° and evaporate it (18)

requires (1178 − 60) × 62·32 = 69674 units of heat, requiring 69674 ÷ 8085 = 8·62 lbs. of average Newcastle coal per nominal horse-power, or 8·62 ÷ 1·5 = 5·75 lbs. per indicated horse-power, &c.

TABLE 43.—Of the HEATING POWER of COMBUSTIBLES, from Cases in Practice.

Kind of Fuel.	Units of Heat or Pounds of Water heated 1° per 1 lb. of Fuel.	Pounds of Water at 212° to Steam at any Pressure per lb. of Fuel.	Pounds of Fuel to evaporate 1 cubic foot of Water at 60° to Steam.	Authority.
Welsh Coals, max. ..	10384	10·75	6·71	H.M. Commissioners.
" min. ..	6143	6·36	11·34	" "
" 37 mean	8742	9·05	7·97	" "
Newcastle, max. ..	9612	9·95	7·25	" "
" min. ..	6559	6·79	10·62	" "
" 18 mean ..	8085	8·37	8·62	" "
Lancashire, max. ..	9148	9·47	7·62	" "
" min. ..	6105	6·32	11·41	" "
" 28 mean	7670	7·94	9·08	" "
Scotch, max.	8172	8·46	8·53	" "
" min.	6839	7·08	10·19	" "
" 8 mean ..	7438	7·70	9·37	" "
Derbyshire, max. ..	8230	8·52	8·47	" "
" min. ..	6105	6·32	11·41	" "
" 7 mean	7322	7·58	9·52	" "
Duffryn Coals	9220	9·54	7·55	Easton & Amos.
" "	8066	8·35	8·64	"
Welsh	7560	7·83	9·21	" (weekly).
Yorkshire, small ..	6172	6·39	11·29	" "
Wood (·2 Water) ..	5494	5·68	12·68	Péclet.
" (·25 Water) ..	3500	3·62	19·90	"
Peat	2400	2·48	29·03	Tredgold.
	(1)	(2)	(3)	

(75.) "*Air required to support Combustion.*"—A knowledge of the quantity of air necessary for different combustibles is important, in order to determine the area of flues, &c. As we have seen in (57) the air has to supply the oxygen necessary for transforming carbon into carbonic acid, and hydrogen into water. Carbonic acid being composed of 1 atom or 75 of carbon, and 2 atoms or 200 of oxygen, a pound of carbonic

acid consists of $75 \div (75 + 200) = \cdot 2727$ carbon and $200 \div (75 + 200) = \cdot 7273$ oxygen; a pound of carbon will therefore require $\cdot 7273 \div \cdot 2727 = 2 \cdot 67$ lbs. of oxygen. Atmospheric air is composed of 2 atoms, or by Table 40, 350 of nitrogen, and 1 atom (100) of oxygen: it follows that 1 lb. of air contains $100 \div (350 + 100) = \cdot 222$ lb. of oxygen, and to yield $2 \cdot 67$ lbs. we require $2 \cdot 67 \div \cdot 222 = 12 \cdot 03$ lbs. of air, or by Table 24, $12 \cdot 03 \div \cdot 0761 = 158$ cubic feet of air at 62°, which is the minimum amount necessary for the combustion of a pound of carbon, the *whole* of the oxygen in the air being consumed (76).

Similarly for hydrogen, we find that water, consisting of 1 atom of oxygen (100) and 1 atom of hydrogen ($12 \cdot 5$), 1 lb. of water consists of $12 \cdot 5 \div (100 + 12 \cdot 5) = \cdot 111$ hydrogen, and $100 \div (100 + 12 \cdot 5) = \cdot 889$ oxygen. One pound of hydrogen requires therefore $\cdot 889 \div \cdot 111 = 8$ lbs. of oxygen, which is the amount contained in $8 \div \cdot 222 = 36$ lbs. or $36 \div \cdot 0761 = 473$ cubic feet of common air at 62°, and this is the minimum amount necessary for the combustion of a pound of hydrogen.

(76.) The quantities of air as found by the preceding calculations are as stated the *minima* absolutely necessary to furnish the oxygen required to support combustion. Practice has led to the use of much larger quantities, the principal reason being perhaps to avoid the formation of carbonic *oxide* instead of carbonic *acid*, which would be the case if the supply of oxygen were too small, the result being a great loss of useful effect, as shown by (109), &c. Analyses of the air that has passed through the fires of well-arranged steam-boilers show that the air still retains half the normal amount of oxygen, and that double the minimum quantity has been used; and we may admit as a practical rule that the quantity of air should be double the minimum theoretical quantity.

(77.) From these data we can easily calculate the quantity of air required for any combustible whose composition is known: thus the average elements of coal are given by Table 41 at $\cdot 804$ carbon and $\cdot 0519$ hydrogen, the last is in (60) reduced to $\cdot 04206$ hydrogen *in excess*, and we shall require

$\left(\cdot 804 \times 158) + (\cdot 04206 \times 473\right) \times 2 = 294$ cubic feet of air at 62° per pound of coal. Calculating in this way with the data in (60), &c., we obtain the following results:—

		Cubic feet of Air at 62°.
Coals	$\left(\cdot 804 \times 158) + (\cdot 04216 \times 473\right) \times 2 =$	294
Peat, dry	$\left(\cdot 58 \times 158) + (\cdot 02 \times 473\right) \times 2 =$	202
„ ordinary state	$\left(\cdot 464 \times 158) + (\cdot 017 \times 473\right) \times 2 =$	163
Coke	$\left(\cdot 85 \times 158\right) \times 2 =$	269
Wood, dry	$\left(\cdot 51 \times 158\right) \times 2 =$	161
„ ordinary state	$\left(\cdot 408 \times 158\right) \times 2 =$	129
Charcoal	$\left(\cdot 93 \times 158\right) \times 2 =$	294

THE VOLUME OF GAS, ETC., PRODUCED BY THE DIFFERENT COMBUSTIBLES.

(78.) The volume of the gases and vapours after combustion depends of course very much on the temperature at which they are taken, but it will be convenient first to consider them at 62°, or the temperature at which we supposed them to enter the fire; the true volume at the actual temperature can then be easily calculated.

Taking first, combustibles containing carbon only, which is the case with coke and charcoal, we find by (75) that 1 lb. of carbon combining with 2·67 lbs. of oxygen forms 3·67 lbs. of carbonic acid, the volume of which by Table 39 is $3{\cdot}67 \times 8{\cdot}59 = 31{\cdot}52$ cubic feet. It is remarkable that this is almost precisely the volume of the oxygen alone, which is $2{\cdot}67 \times 11{\cdot}88 = 31{\cdot}72$ cubic feet, so that when oxygen and carbon combine, the volume of the carbonic acid gas formed is nearly the same as that of the oxygen consumed: when, therefore, a combustible contains carbon only, the volume of gas in the chimney is the same as that of the air entering the fire, expanded of course to the volume due to the increased tempera-

ture, the oxygen consumed having been replaced by the same volume of carbonic acid gas. The nitrogen in the air is passive, passing through the fire without chemical alteration.

(79.) When a combustible contains hydrogen, it combines with eight times its own weight of oxygen, derived either from the fuel itself or from the air which supports the combustion; in either case water is formed which again forms vapour. If the combustible contains water already formed, and with which it is more or less saturated, vapour is formed from it, and is added to the products of combustion.

Table 39 gives the volume of 1 lb. of vapour of water at 62°, and a pressure of 29·92 inches of mercury, at 21·07 cubic feet; this is fictitious, as stated in (55), for as shown by col. 4 of Table 68, vapour at 62° can have none other elastic force than ·556 inch of mercury, and 1 lb. of vapour at that pressure will occupy by col. 11 of the same table, 1135 cubic feet, still the effect of 1 lb. of vapour in a large volume of air is to increase the volume of that air 21 cubic feet, whether it saturates it or not. Say, that we have a vessel containing 1135 cubic feet of *dry* air at 62° and at 30 inches of mercury in the barometer; a pound of water would become vapour, and just suffice to saturate it, becoming thus 1135 cubic feet of vapour at 62°. But the pressure would be the sum of the two pressures before mixture, it would therefore become 30 + ·556 = 30·556 inches; by enlarging the vessel we can reduce that pressure to 30 inches again, and the volume required for that is = 1135 × 30·556 ÷ 30 = 1156 cubic feet. The volume of air has therefore been increased 1156 − 1135 = 21 cubic feet, by the addition of 1 lb. of water converted to vapour, agreeing thus with col. 5 of Table 39. If the volume of air had been greater, the pound of water would not have saturated it, but the increase of volume would have been the same; thus if the volume of dry air had been ten times greater, or 11,350 cubic feet, the pound of water would have filled it with vapour, but the tension of that vapour would have been one-tenth only of the maximum tension, or ·0556 inch of mercury; the pressure of the mixture then becomes 30 + ·0556 = 30·0556 inches, and to reduce it to 30 inches, the volume must be increased to 11350 × 30·0556 ÷

30 = 11371 cubic feet, being an increase of 11371 − 11350 = 21 cubic feet by 1 lb. of vapour as before. We can now easily calculate the volume of vapour at 62° formed by the combustion of a pound of fuel whose composition is known.

Thus 1 lb. of coal, by Table 41 contains ·0519 lb. of hydrogen, which combining with $\cdot 0519 \times 8 = \cdot 4152$ lb. of oxygen forms $\cdot 0519 + \cdot 4152 = \cdot 4671$ lb. of water, which again by Table 39 forms $\cdot 4671 \times 21 \cdot 07 = 9 \cdot 84$ cubic feet of vapour reduced to 62°.

Wood perfectly dry contains $\left(\cdot 053 + (\cdot 053 \times 8\right) \times 21 \cdot 07 =$ 10·05 cubic feet of vapour at 62°.

Wood in the ordinary state of dryness gives out $\left(\cdot 042 + (\cdot 042 \times 8\right) = \cdot 378$ lb. of vapour, which added to the ·2 water already formed in it, gives a total of $(\cdot 378 + \cdot 2) \times 21 \cdot 07 = 12 \cdot 1$ cubic feet of vapour at 62°.

Peat perfectly dry gives $\left(\cdot 06 + (\cdot 06 \times 8\right) \times 21 \cdot 07 = 11 \cdot 38$ cubic feet of vapour at 62°.

Peat in the ordinary state gives $\left(\cdot 048 + (\cdot 048 \times 8\right) = \cdot 432$ lb. of water, which added to the ·2 water already formed in it gives a total of $(\cdot 432 + \cdot 2) \times 21 \cdot 07 = 13 \cdot 3$ cubic feet of vapour at 62°.

(80.) In most cases the temperature of the air, &c., in the chimneys of steam-boilers is about 550° (100), at which the volume of air by Table 24 is double the volume at 62°. Admitting this temperature, and collecting from (77) and (79) the data there obtained, we get the volumes of air, gases, and vapour at 550° in the chimney as follows:—

Coal	$(294 + 9 \cdot 84) \times 2 = 608$	cubic feet at 550° per lb. of fuel.
Wood, perfectly dry	$(161 + 10 \cdot 05) \times 2 = 342$	„ „
„ ordinary state	$(129 + 12 \cdot 1\ ) \times 2 = 282$	„ „
Peat, perfectly dry ..	$(202 + 11 \cdot 38) \times 2 = 427$	„ „
„ ordinary state	$(163 + 13 \cdot 3\ ) \times 2 = 352$	„ „
Coke	$269 \times 2 = 538$	„ „
Charcoal	$294 \times 2 = 588$	„ „

ON THE MODES IN WHICH COMBUSTIBLES YIELD THEIR HEAT UNDER DIFFERENT CIRCUMSTANCES, ETC.

(81.) To illustrate the way in which combustibles give out their heat, we will take the case shown by Fig. 1 of a ball heated to say 600°, and placed in a room or vessel whose walls are maintained at 100°, and let the space enclosed be a vacuum. Then the ball will give out its heat to the walls entirely by *radiation*, and will continue to do so till it is reduced to the same temperature as those walls.

(82.) Now take another case, represented by Fig. 2, in which a ball heated to 600° is placed in a room, whose walls have the same temperature as itself; and let air at say 100° be admitted at A and allowed to escape at B, becoming heated in its passage by contact with the ball and the walls, and cooling them continuously and simultaneously, until both are reduced to its own temperature. Here we have a case in which all the heat of the ball is given out *by contact of cold air*, and none by radiation, because the walls have throughout the same temperature as the ball.

(83.) To complete our illustrations, let Fig. 3 represent a case in which the heat of the ball is carried off by *both* causes. Say the ball is heated to 600°, while the walls are at 300° and the air at 100°; here part of the heat will be given out by radiation to the walls, as in our first illustration, and part to the cold air passing through the room, as in our second illustration.

Applying all this to combustibles, we find the case shown by Fig. 1 and (81) impossible, inasmuch as air is absolutely necessary to support combustion; but the other two cases are practicable.

(84.) "*Temperature of the Air, &c., from Furnaces.*"—Let Fig. 4 be a furnace constructed of fire-brick, which for the purpose of illustration we may suppose to be a perfect non-conductor of heat, so that its interior surface will have the same temperature as the fire, and we have a case analogous to Fig. 2 (82); no heat will be given out by radiation, but the whole of the caloric yielded by the fuel will be carried off

TABLE 44.—Of the HEATING POWER of COMBUSTIBLES, the quantity of Air required for the Combustion of One Pound of Fuel, and the Quantity and Temperature of the Air, &c., in the Chimney.

KIND OF FUEL.	Total Heat in a Pound.	Heat radiated.		Heat given out to Air supporting Combustion.		Air used.		Cubic Feet of Vapour at 62°.	Cubic Feet, Air and Vapour in Chimney at 550°.	Temperature of Air as it leaves the Fire.	
	Units.	Per cent.	Units.	Per cent.	Units.	Lbs.	Cubic Feet at 62°.			Steam-boiler with internal Fire.	Fire-brick Furnace.
										°	°
Coals	13000	50	6500	50	6500	22·4	294	9·84	608	1282	2256
Coke	10970	50	5485	50	5485	20·47	269	..	538	1188	2089
Wood, artificially dried	6582	23	1514	77	5068	12·25	161	10·05	342	1869	2173
„ ordinary state (·2 water)	5265	23	1211	77	4054	9·81	129	12·1	282	1797	2091
„ Charcoal	12000	45	5400	55	6600	22·4	294	..	588	1326	2130
Peat, artificially dried	8736	50	4368	50	4368	15·38	202	11·38	427	1256	2211
„ ordinary state (·2 water)	7151	50	3575	50	3575	12·4	163	13·8	352	1274	2244
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)

by the air supporting the combustion. This air will be heated to a temperature varying with the volume admitted and the heating power of the fuel.

(85.) Thus by (61) charcoal yields 12,000 units per pound, and by (77) requires 294 cubic feet or $294 \times \cdot 0761 = 22 \cdot 4$ lbs. of air at 62°. A pound of charcoal will therefore heat a pound of water 12,000°, and the specific heat of air (5) being ·238, it would heat a pound of air $12000 \div \cdot 238 = 50420°$, and as we have 22·4 lbs. of air to carry off that heat, the increase of its temperature will be $50420 \div 22 \cdot 4 = 2251°$, and as it enters the fire at 62°, it will depart at $2251 + 62° = 2313°$.

(86.) This, however, is not strictly correct, as it rests on the assumption that the air passes through the fire unchanged in weight and specific heat, which is not a fact: thus 1 lb. of charcoal contains ·93 lb. of carbon, which combining with $\cdot 93 \times 2 \cdot 67 = 2 \cdot 483$ lbs. of oxygen forms $\cdot 93 + 2 \cdot 483 = 3 \cdot 413$ lbs. of carbonic acid gas, to heat which 1° requires $3 \cdot 413 \times \cdot 2164 = \cdot 738$ unit of heat. The oxygen being derived from the air, which is composed of ·222 oxygen and ·778 nitrogen, the 2·483 lbs. of oxygen is combined with $2 \cdot 483 \times \cdot 778 \div \cdot 222 = 8 \cdot 7$ lbs. of nitrogen, to heat which 1° requires $8 \cdot 7 \times \cdot 244 = 2 \cdot 1228$ units: hence to heat the products of the combustion of a pound of charcoal 1°, we require $\cdot 738 + 2 \cdot 1228 = 2 \cdot 8608$ units, and as the combustion developes 12,000 units, the temperature will be raised $12000 \div 2 \cdot 8608 = 4194°$ or to $4194 + 62 = 4256°$ when the whole of the oxygen in the air is consumed (76), but when only half of it is consumed, which we have fixed as a practical condition, the increase of temperature becomes $4194 \div 2 = 2097°$, and the final temperature $2097 + 62 = 2159°$, instead of 2313° as in (85). The difference is not great, and may be neglected for practical purposes.

(87.) We assumed for the purpose of illustration, that the fire-brick of which the furnace was constructed, was a perfect non-conductor, which is not a fact; a considerable portion of the heat will be transmitted to the outer surface and dissipated by radiation there. Allowing that 10 per cent. is dissipated by radiation, &c., in the case Fig. 4, the heat carried off by the air is reduced to $12000 \times \cdot 9 = 10800$ units and the exit temperature

to 10800 $\div$ (22·4 $\times$ ·238) + 62° = 2088°. The other kinds of fuel yielding their respective quantities of heat to the air required for their combustion, will give different temperatures to that air, as per col. 11 of Table 44, allowing throughout 10 per cent. for loss by radiation. Thus with coals, we require by (77) 294 cubic feet or 294 $\times$ ·0761 = 22·4 lbs. of air, which having to carry 13000 $\times$ ·9 = 11700 units, will be heated 11700 $\div$ (22·4 $\times$ ·238) = 2194°, and entering the fire at 62°, will depart at 62 + 2194 = 2256°.

The temperature of the air will also vary with the volume allowed for combustion. Table 45 gives the variation of temperature for coals with different volumes of air.

TABLE 45.—Of the TEMPERATURE of the AIR from a FIRE-BRICK FURNACE, showing the effect of using different volumes of Air.

State of the Oxygen.	Cubic Feet.		Lbs.	Increase of Temp.		Temp. of Atmosphere.		Temp. of Air as it leaves the Fire.
				°		°		°
Half burnt	294	=	22·4	2194	+	62	=	2256
Quarter burnt ..	588	=	44·8	1097	+	62	=	1159
One-fifth burnt ..	735	=	56·00	878	+	62	=	940

(88.) "*Means of obtaining very high Temperatures.*"—We have seen in (87) and by Table 45, that the temperature is increased by reducing the volume of air; if the whole of the oxygen in the air were consumed, we should require only 11·2 lbs. of air per pound of coal, and the temperature at exit would become 11700 $\div$ (·238 $\times$ 11·2) + 62° = 4451°. But to obtain this result, special precautions would be necessary to avoid the formation of carbonic *oxide* instead of carbonic acid: the fuel would require to be in considerable thickness, so that the air had a considerable distance to travel through the fire, and the air would required to be supplied by the blast of bellows, &c. With charcoal we found in (86) the exact temperature to be 4256° when the whole of the oxygen in the air was consumed; with pure oxygen a most intense heat would be obtained, namely, (12000 $\div$ ·738) + 62 = 16322°, but the costliness of oxygen will prevent its application to practice.

(89.) We will now investigate the phenomena of combustion in cases analogous to Fig. 3, where the fuel is more or less surrounded by surfaces of low temperature which absorb the radiant heat. By studying our three illustrations, it will be evident that the amount radiated will vary with the temperature of the absorbing surface, for if, as in Fig. 2, the walls were of the same temperature as the fire, no radiant heat would be given out or received, and generally the lower the temperature of the absorbent, the more radiant heat would be received by it.

THE RADIATING POWER OF COMBUSTIBLES.

(90.) Péclet's apparatus for ascertaining the radiating power of combustibles is shown in Fig. 5: it consisted of an annular vessel of tin plate, the interval between the two cylinders being filled with water, the temperature of which was given by two thermometers with long bulbs, whose stems pass through corks. The internal cylinder was open at both ends, its surface was coated with lamp-black, and in its centre was suspended a cage of wire containing the fuel. In using this apparatus a given weight of fuel in a state of ignition is introduced and consumed, the radiant heat and that alone is absorbed, and raises the temperature of the water and the vessel itself: knowing the weight of the water, and of the vessel, also the increase of temperature with a given weight of fuel, we can calculate the heat given out by radiation. But a correction is necessary here, for evidently with this form of apparatus we only obtain *part* of the heat radiated, because radiation takes place equally in *all directions*, and part of it must escape by the open ends of the cylinder, and be lost. In Péclet's apparatus the internal cylinder was 8 inches diameter and 12 inches high, and the ratio of the total radiant heat to the portion absorbed was in that case as 1·2 to 1, as shown by the following investigation.

(91.) The principles on which this correction is made are illustrated by Fig. 6, in which A is the radiant body, say at 600° placed in the centre of a hollow sphere, C D E F, at 100°. The body, A, will send out radiant heat equally in all directions, which will be absorbed by the sphere, but if two segments, C D

and E F, be cut out and removed, the radiant heat that would have fallen on them passes out into the atmosphere and is lost; but knowing the proportion which those two segments bear to the whole sphere, we can estimate the amount lost by their removal. The rules of mensuration show that the surface of any sphere or segment of a sphere, is given by multiplying the circumference of the sphere by the diameter, or by the height of the segment. The distances C E and C D are given, in our case 12 in. and 8 in. respectively, and the angle C E F being a right angle, we get the diameter C F of the circumscribing sphere $= \sqrt{12^2 + 8^2} = 14 \cdot 4$ inches or $45 \cdot 2$ inches circumference, and $45 \cdot 2 \times 14 \cdot 4 = 650$ total surface. The area of the two segments is $45 \cdot 2 \times 1 \cdot 2 \times 2 = 108$, leaving in the apparatus $650 - 108 = 542$ square inches to absorb the heat, the ratio of the surface of the whole sphere to which is $650 \div 542 = 1 \cdot 2$ to 1.

(92.) With this apparatus the weight of water was $23 \cdot 84$ lbs., and the vessel itself which was made of tin plate weighed $4 \cdot 9$ lbs. The combustion of $\cdot 1232$ lb. of wood charcoal raised the temperature $25^\circ \cdot 2$, the water received $23 \cdot 84 \times 25 \cdot 2 = 600 \cdot 7$ units of heat, and the vessel whose specific heat (2) was $\cdot 11$ received $4 \cdot 9 \times 25 \cdot 2 \times \cdot 11 = 13 \cdot 6$ units, altogether $600 \cdot 7 + 13 \cdot 6 = 614 \cdot 3$ units, which with a completely surrounding surface would have been $614 \cdot 3 \times 1 \cdot 2 = 737$ units, or $737 \div \cdot 1232 = 5983$ units per pound of fuel, and the total heating power being as we have shown (61) 12,000 units, the radiant heat is $5983 \div 12000 = \cdot 5$ nearly, so that 50 per cent. is given out by radiation, and 50 per cent. to the air passing through the fuel.

(93.) The temperature of the ignited fuel would be about 2200°, and the *mean* temperature of the absorbing surface being 70° in the above experiment, the difference of temperature was $2200 - 70 = 2130^\circ$, whereas if the absorbing surface had been at 300°, as in high-pressure steam-boilers, the difference would have been $2200 - 300 = 1900^\circ$ and the heat lost by radiation would have been $(5983 \times 1900) \div 2130 = 5337$ units, which is $5337 \div 12000 = \cdot 445$, or say 45 per cent. of the heat in the fuel, leaving 55 per cent. to pass off in the air.

(94.) In another experiment with oak wood, made with the same apparatus, $\cdot 2145$ lb. of wood raised $23\cdot 84$ lbs. of water and $4\cdot 9$ lbs. of tin plate 9°, which is equal to

$$\frac{\left(23\cdot 84 + (4\cdot 9 \times \cdot 11\right) \times 9 \times 1\cdot 2}{\cdot 2145} = 1228 \text{ units per pound,}$$

and the total heating power of ordinary wood being (61) as we have shown 5265 units, the radiant heat is $1228 \div 5265 = \cdot 233$—say 23 per cent.; leaving 77 per cent. to pass off with the air.

(95.) In another experiment with peat charcoal, by the combustion of $\cdot 0858$ lb. of fuel the temperature was raised $15^\circ\cdot 3$, which is equal to $\frac{\left(23\cdot 84 + (4\cdot 9 \times \cdot 11\right) \times 15\cdot 3 \times 1\cdot 2}{\cdot 0858} = 5217$ units per pound, and as the total heating power is by (62) 10,557 units, the radiant heat is $5217 \div 10557 = \cdot 5$ nearly, or 50 per cent. of the total heat; leaving 50 per cent. to heat the air. Péclet estimates the radiant power of peat itself to be 50 per cent. also, the same as peat charcoal.

(96.) The radiant power of coal, coke, &c., could not be satisfactorily determined by this apparatus, from the difficulty of keeping a small quantity of fuel ignited; but by comparison Péclet estimates them as rather more than wood charcoal, which we found to be 45 per cent. (93); we may therefore estimate the radiant power of coal and coke at 50 per cent.

(97.) We have shown (85) that the air leaving a fire of charcoal, when *all* loss by radiation was avoided, had the temperature of 2313°; but when 10 per cent. was radiated (87) the temperature was reduced to 2088°. When the fire is surrounded by cool surfaces 50 per cent. is radiated, and the temperature of the air is greatly reduced, the same weight having to receive only half the amount of heat. In the case of coals, the air leaves the fire at $\frac{13000 \times \cdot 5}{22\cdot 4 \times \cdot 238} + 62^\circ = 1282^\circ$; with wood in the ordinary state of dryness, $\frac{5265 \times \cdot 77}{9\cdot 817 \times \cdot 238} + 62^\circ = 1797^\circ$; and so on with the

rest, as in col. 10 of Table 44, which gives a general and collected statement of the facts, arrived at in the foregoing investigation.

(98.) "*Combustion with Steam-boilers.*"—We may now apply all this to the case of a steam-boiler, and will assume that with ordinary firing about 5 per cent. more of the combustible matter in the fuel falls unconsumed from the grate, than in the experiments from which the data are derived, the useful heat being reduced to $13000 \times \cdot 95 = 12350$ units; we will also assume that each pound of coals requires 300 cubic feet, or 22·83 lbs. of air at 62°.

Let Fig. 7 be a boiler or fire-box—not of a practical form, but such as to serve for the illustration of the case in which the fire being completely surrounded by an absorbing surface, will *lose* none of its radiant heat. In such a case 50 per cent. of the total heat in the fuel will be given out by radiation, and 50 per cent. to the air by which the combustion is supported; the first, or 6175 units, is absorbed by the side, &c., of the fire-box, and 6175 units pass off with the air into the chimney and are lost. The temperature of the air as it departs may be found as before; the heat carried off by it would heat 1 lb. of water 6175°, therefore a pound of air, $6175 \div \cdot 238 = 25950°$, and as we have 22·83 lbs. of air the increase of temperature will be $25950 \div 22 \cdot 83 = 1138°$, and the final temperature $1138° + 62° = 1200°$.

(99.) This temperature will vary with the quantity of air admitted, as is shown by Table 46.

TABLE 46.—Of the EFFECT of DIFFERENT VOLUMES of AIR in STEAM-BOILER FURNACES.

State of the Oxygen.	Cubic Feet.		Lbs.	Increase of Temp.		Temp. of Atmosphere.		Temp. of Air as it leaves the Fire.
				°		°		°
Half burnt	300	=	22·83	1138	+	62	=	1200
Quarter burnt ..	600	=	45·66	568	+	62	=	630
One-fifth burnt ..	750	=	57·00	455	+	62	=	517

(100.) We have seen that a boiler, like Fig. 7, in which the air escapes immediately out of the furnace, 50 per cent. of the

heat in the fuel is lost. Part of this may be recovered by causing the heated air to pass in contact with the surface of the boiler, through a long circuit on its way to the chimney. In Fig. 8 we have a fire-box as in Fig. 7, and, for the purpose of illustration, we will suppose the boiler to have no return flue, but to be of great length, with the chimney at the end. It is found by observation of well-arranged boilers, that the air passes into the chimney at about 550°,—a high temperature is necessary to obtain the proper draught, as will appear hereafter (174); we have seen that as it leaves the fire-box it is at 1200°, while from end to end the absorbing surface in the case of a high-pressure boiler is at 300°.

(101.) The amount of heat abstracted from the air at each point will be in proportion to the *difference* between the temperature of the air at that point and the temperature of the boiler; at the fire-box end the difference is $1200° - 300 = 900°$, while at the chimney end it is only $550° - 300° = 250°$. Say we assume that one-tenth of this difference of temperature is parted with from point to point throughout, then the first difference being 900°, one-tenth of that is 90°, and the next point will have a temperature of $1200° - 90° = 1110°$. The difference is now, therefore, $1110° - 300° = 810°$, and the temperature of the second point will be $1110° - \frac{810}{10} = 1029°$, and thus we have calculated the successive temperatures in Fig. 8.

(102.) The air departing into the chimney at the high temperature of 552°, carries off a considerable amount of heat, namely, the amount required to heat 22·83 lbs. of air 490° (or $552° - 62°$); and this is equal to $22{\cdot}83 \times 490 \times {\cdot}238 = 2663$ units, or about 20 per cent. of the total heat in the coals.

(103.) There is also another source of loss of heat, namely, by radiation and contact of cold air with the outside of the boiler and its brickwork, whereby the boiler-house is kept at a high temperature. The loss from this cause will vary very greatly with the greater or less exposure, &c.; but for ordinary Cornish boilers, set in brickwork in the usual way in a closed boiler-

house, we may take this loss at 12 per cent. Collecting these results, we have the total heat in 1 lb. of coal (13,000 units) distributed as shown by Table 47.

TABLE 47.—Of the DISTRIBUTION of the HEAT in ONE POUND of COALS, by an ordinary Boiler, with internal Fire.

In Ashes, left unburnt	13000 × 5% =	650 units
Lost by Air in Chimney	13000 × 20% =	2600 ,,
Lost by Radiation, &c., in Boiler-house ..	13000 × 12% =	1560 ,,
Utilized in production of Steam	13000 × 63% =	8190 ,,
	Total	13000 ,,

(104.) "*Efficiency of Long and Short Boilers.*"—We have stated (100) that a high temperature of the air in the chimney is necessary in order to obtain a good draught; but if it were not so, an extreme length of boiler would be necessary to secure even a small portion of the heat wasted. Say we doubled the length of our boiler, Fig. 8, continuing the calculation of the successive temperatures, we obtain the series given by Fig. 9. The heat lost by the chimney in this case would be the amount necessary to heat 22·83 lbs. of air 307° (or 369° − 62°), or $22{\cdot}83 \times 307 \times {\cdot}238 = 1668$ units. With the ordinary length of boiler the loss from the same cause was 2663 units (102); so that by *doubling* the length we obtain $2663 - 1668 = 995$ units only, or 12 per cent. increase on the useful effect (8190 units).

(105.) But even this would not be realized. In Table 47 we have allowed 1560 units as the loss in the boiler-house, &c., by a boiler of ordinary length: with double the length, the loss from the same cause would be greater, and would dissipate much of the extra heat obtained by the extra length. Indeed, it is obvious that beyond a certain point we should, by increasing the length, lose more from one cause than we should gain from the other. If we assume that in a boiler of ordinary length one-half of the loss by radiation, &c., is due to the exposed front and furnace, and the other half to the body of the boiler, we shall be conducted to the results shown by Table 48.

TABLE 48.—Of the HEAT LOST by LONG and SHORT BOILERS, &c.

Proportional Length of Boiler.	Temperature of Air in the Chimney.	Heat lost by Air in Chimney.	Heat lost by Radiation, &c. By Front, &c.	By Body.	Total.	Left in the Ashes.	Heat utilized.	Ratio of Economy.	Gain or Loss per cent.
	°	units.	units.	units.	units.	units.	units.		
$\frac{1}{2}$	778	3890	780 +	390 =	1170	650	7290	897	−10·3
$\frac{3}{4}$	648	3184	780 +	585 =	1365	650	7802	960	− 4·0
1	552	2663	780 +	780 =	1560	650	8127	1000	0·0
$1\frac{1}{2}$	432	2010	780 +	1170 =	1950	650	8390	1032	+ 3·2
2	369	1668	780 +	1560 =	2340	650	8342	1026	+ 2·6

This would show that by reducing the boiler to half the usual length we should only lose 10 per cent.; one, half as long again as usual, would give only 3·2 per cent. more useful heat; and by increasing the length to double we should actually *lose* rather than gain. These results are not given as absolutely correct, but will at least serve to show approximately the relative effect of long and short boilers.

(106.) We have so far supposed that a certain fixed quantity of fuel was used with a certain fixed volume of air, but with a powerful chimney and a regulated draught, the velocity of the current of air might be so adjusted that the volume was greater or less at pleasure. Say we admit a double volume, producing in our case a double velocity: if a double quantity of coals be burnt in the *same time*, we should have the common case of a fire forced beyond its proper intensity: if, on the other hand, the quantity of fuel was not increased, then twice the necessary quantity of air would be used. In both cases a loss of useful effect would ensue, the amount of which we will proceed to investigate.

(107.) "*Effect of Forced Firing.*"—Taking the first case of a forced fire, and referring again to Fig. 8, we shall observe that with double velocity, the air which in a given time moved half the length of the boiler from A to B, will in our new case have moved from A to C; and as the amount of heat lost by it is *a question of time*, it follows that instead of being reduced to 552°, as at C, it will be reduced to 778° only, as at B, and will pass off into the chimney at this last temperature, bearing off with it 778° − 552° = 226° more heat than in the former case; so that

by forced firing we lose for each pound of coal $226 \times 22 \cdot 83 \times \cdot 238 = 1228$ units of heat; and as with ordinary firing we obtained 8190 units (Table 47), we now obtain only $8190 - 1228 = 6962$ units, which is $6962 \div 8190 = \cdot 85$, or 85 per cent., showing a loss of 15 per cent. by forcing the fire to the extent of a double consumption of fuel.

(108.) "*Effect of too much Air.*"—When more air is used than is necessary to effect the proper combustion, a great loss occurs. We shall have as before 6175 units given out by radiation to the fire-box surrounding the fuel, and 6175 units to be carried off by the air; but, instead of 22·83 lbs. of air per pound of coal, we shall now have 45·66 lbs. of air; its temperature will therefore be raised $\frac{6175}{45 \cdot 66 \times \cdot 238} = 569°$, and the atmospheric temperature being 62°, it will leave the fire at $569° + 62° = 631°$, instead of 1200°. Then, calculating as before with Fig. 8 and (101), we have $631° - 300° = 331°$ for the first difference, and $331 \div 10 = 33°$ for the decrease between the first and second points; the second point will therefore be $631° - 33° = 598°$, &c., &c.; and thus we obtain the series of numbers in Fig. 10. It will be observed that as the velocity is double, the distances between point and point will be doubled, and the air will pass into the chimney at 475°, or $475° - 62° = 413°$ above the temperature of the atmosphere; and the amount of heat lost thereby will be $45 \cdot 66 \times 413 \times \cdot 238 = 4488$ units, and we have—

In ashes left unburnt	650	units
Lost by air in chimney	4488	,,
Ditto by radiation, &c., in boiler-house ..	1560	,,
Utilized in production of steam	6302	,,
	13000	,,

With the proper quantity of air, 8190 units were utilized per pound of coal (Table 47), whereas now we have only 6302 units, showing a loss of 23 per cent. of useful effect, by the admission of double the necessary quantity of air.

This is a fruitful source of loss of fuel in very many cases: an ignorant stoker delights in a roaring fire and sharp draught,

unconscious of the loss of fuel incurred; in all cases the damper should be regulated so as to produce a moderate draught, and this is especially important where there is a tall or powerful chimney.

(109.) "*Effect of too little Air.*"—Care must be taken on the other hand not to curtail the supply of air too much, as in that case also a great loss would arise from the formation of carbonic *oxide* instead of carbonic *acid;* carbonic oxide being formed of one equivalent of carbon and one oxygen, whereas carbonic acid is formed of carbon 1 and oxygen 2.

The experiments of Favre and Silbermann in Table 42 show that a pound of carbon burning to carbonic oxide yields only 4453 units of heat, whereas it would yield 12,906 units in burning to carbonic acid by the experiments of Dulong. The coal we have been considering (60), which is composed of ·812 carbon and ·04206 hydrogen *in excess*, will give—

In carbon burning to carbonic oxide	$\cdot 804 \times 4453 =$	3580 units
„ hydrogen in excess to water	$\cdot 04206 \times 62535 =$	2630 „
		6210 „

The same coal with a proper quantity of oxygen (60) gave 13,000 units, we have therefore in our case only $6210 \div 13000 = \cdot 478$, or say 48 per cent. of the available heat which the fuel could supply.

(110.) We have here taken an extreme case, the quantity of air being only half the amount absolutely necessary for proper combustion, and by (76) one-fourth of the amount usually consumed in well-regulated furnaces. Where the air is curtailed, but to a less degree, part of the carbon will be transformed into carbonic acid, and part into carbonic oxide, and the result will be intermediate between the extremes we have given. Say we have $1\frac{1}{2}$ oxygen to 1 carbon; in that case they will still combine only in the proper proportions to form one or other of the two products, that is to say, 1 to 1, or 1 to 2, but they may arrange themselves thus: we have $1\frac{1}{2}$ oxygen which will divide itself into two unequal portions, 1 oxygen combining with $\frac{1}{2}$ carbon, forming $1\frac{1}{2}$ carbonic acid, and $\frac{1}{2}$ oxygen combining with $\frac{1}{2}$ carbon,

forming 1 carbonic oxide, and in that case the coals we have considered would give—

In carbon burning to carbonic acid	·402	× 12906	=	5188	units
„ „ „ oxide	·402	× 4453	=	1790	„
„ hydrogen „ water	·04206	× 62535	=	2630	„
				9608	„

In this case we should have therefore 9608 ÷ 13000 = ·74, or 74 per cent. of the heat due to the best condition with the proper quantity of air.

(111.) This will explain the anomalous fact that where there is a bad draught, not only is there difficulty in keeping up the steam, &c., but that there is a great consumption of fuel for the work done; it might be expected that with a slow dull fire few coals would be burnt, or if by dint of forcing, fuel was largely consumed, it *must* yield the heat due to it, but it will be evident that it is possible for there to be a dull fire, a large consumption of fuel, and little useful result at one and the same time, and all this arising from insufficient draught.

It will also be seen that in every case the proper regulation of the damper is a matter of extreme importance, and that nice adjustment is necessary to produce the best effect, too much or too little air causing a great loss of fuel; an intelligent stoker, without any knowledge of the theory, finds by experience the height of damper with which he can do the work with least fuel; if the work varies he watches and adjusts the damper accordingly, and such a man should have more consideration and better wages than he usually receives.

(112.) The general result of our investigation is that the amount of heat utilized in practice varies very much with the size of the boiler (105) and other circumstances, but in ordinary cases we may admit that with large boilers, 8000 units are utilized per pound of coal (74), but with *very* small boilers such as are commonly used for hot-house and similar purposes, not more than 4000 units can be reckoned on: for medium sizes we may allow 6000 units per pound of coal.

As applied to heating air for ventilation, &c., as much as 12,000 units per pound of coal may be utilized (383), (388).

CHAPTER III.

ON STEAM-BOILERS.

HAVING in the preceding chapter investigated the phenomena of combustion, &c., we may apply the results to steam-boilers in practice, checking and, if needs be, modifying our deductions by the dictates of experience. We found in (105) that in all cases the most economical size of boiler was a *medium* one, and that a departure therefrom in either direction was followed by a loss of effect, an excessively long and a very short boiler giving less duty for the fuel used than a medium-sized one, properly proportioned to the work to be done.

(113.) "*Effective Heating Surface.*"—When heated air is in contact with a surface much colder than itself, the amount of heat given out is not only a question of time, but also of *position*, of the receiving surface. Let A, Fig. 16, be a square vessel full of cold water, and let heated air pass along the four flues B C D E; the four surfaces F G H J will absorb very different quantities of heat, although they are all of the same area, &c. The surface F will receive the most, for two reasons: the hottest of the heated air will occupy the upper portion of the flue, in immediate contact with the bottom of the boiler; and the water when heated becomes lighter there, and immediately ascends, and is replaced by colder water; and so the heat received is rapidly distributed through the mass of the water. But the surface H is in the worst possible position, for if the water in contact with it is heated, it becomes lighter and remains persistently in contact with the surface, and the heat is carried downwards very slowly as shown by experiment in (244); moreover, such a surface would usually be covered with a layer of fine ashes, which being a bad conductor of heat, would still further retard its transmission; we may therefore admit that the surface H is useless, receiving no heat whatever. The surfaces G and J are in an intermediate position, and we may assume that they absorb a quantity of heat, a mean between F and H, or

$(0+1) \div 2 = \frac{1}{2}$; the whole boiler therefore receives only half the amount of heat that it wonld have received if its whole surface had been as effective as F. We have thus to consider, not only the *real* surface of a boiler, but the *effective* surface, in estimating the result to be obtained, and we shall assume as a standard a flat and horizontal surface with the heated air beneath it, as so much *effective* surface, and shall reduce all other surfaces to that standard.

(114.) In Fig. 17 we have an octagonal boiler which we will suppose to be placed in a hot-air flue and exposed all over. On the same principles as before we find that if the surface F absorbs an amount of heat represented by 1, then B will receive 0 and C, D, E, the respective quantities $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ as in the figure: the sum of whole is 4, whereas if the eight sides of the figure had all been as effective as F, we should have had 8. Here, therefore, as with a square figure, the effective surface is *half* the real surface exposed.

(115.) Let Fig. 18 be a cylindrical boiler filled with water for illustration as before, and we have the effective surface represented by a series of numbers, as in the figure, from 0 at A to 1 at B, &c.; the mean of the whole is half the maximum as before.

But, with cylindrical boilers heated outside, the lower half of the cylinder only is usually made available as in Figs. 19, 27, in which case the effective surfaco (Fig. 18) is $\frac{\frac{1}{2}+1}{2} = \frac{3}{4}$ of the real surface exposed to the heated air. But where the whole surface is exposed, as is the case with the lower tubes of a French or Elephant boiler, the effective surface is half the real, as we have seen.

(116.) In an internal flue as in Fig. 19, the same numbers occur as in Fig. 18, but in inverted order, the most effective surface being at the top, decreasing to 0 at bottom: here the whole surface is exposed and the effective surface is half the real.

(117.) "*Power of Boilers.*"—In determining the area of heating surface necessary for doing a given amount of work we must be guided by practical experience. There is great latitude here, for we have seen by (107) that a boiler nominally of a

certain power may be forced up to $6962 \times 2 \div 8190 = 1 \cdot 70$, or 70 per cent. beyond its normal power with a *double* consumption of fuel.

The area of fire surface should not be simply proportional to the work to be done, for several reasons: with a small boiler the loss by radiation, &c., is much greater than with a large one, the area exposed to cooling influences being much greater in proportion; then with a small boiler the air commonly passes into the chimney at a higher temperature than with a large one, in spite of the large area of flue adopted to prevent it (138); and as applied to steam-engines, a small engine takes more steam in proportion than a large one, the loss by friction, &c., being greater.

When a boiler is applied for working a steam-engine its power is usually estimated by that of the engine which it supplies, and this practice has led to the application of the term "horse-power" to all boilers whether used for engines or not. This is much to be regretted; the amount of steam is not simply proportional to the power of the engines even where they are of precisely the same construction, the loss by friction, &c., being greater in proportion with small engines; moreover, different kinds of engines require very different quantities of steam to do the same work, expansive engines taking less than others. Thus 60 lbs. steam cut off at $\frac{1}{5}$th takes only half the fuel required for steam of the same pressure acting without expansion, and doing the same work.

The term horse-power is so generally used by practical men, that it is hopeless to expect it to be abandoned, and in deference to custom we shall make use of it, first defining what we understand it to mean.

(118.) We shall estimate a cubic foot of water at 60° evaporated to steam at any pressure (19) as equal to 1 nominal horse-power, and this by (18) is equivalent to $(1178 - 60) \times 62 \cdot 32 = 69674$, say 70,000 units of heat. The nett *indicated* horse-power we consider equal to $1\frac{1}{2}$ times the nominal, therefore $1 \div 1\frac{1}{2} = \cdot 667$, or $\frac{2}{3}$rds of a cubic foot of water at 60° to steam is equal to 1 nett indicated horse-power.

It has been found by experience that a 4-horse boiler requires

about 18 *effective* square feet of surface per nominal horse-power; 10-horse, about 14 square feet; 20 horse, about 12; and 50-horse, about 11. The best rule we can give is an empirical one:

$$A = \left(H + (\sqrt{H} \times 2{\cdot}5\right) \times 8,$$

in which H = the nominal horse-power, and A = the effective area of the boiler in square feet estimated as explained in (113), &c. Thus, with the 50-horse boiler, Fig. 36, the body of the boiler having its lower half only exposed to the heated air gives

TABLE 49.—Of the AREA of SURFACE for STEAM-BOILERS and their FIRE-GRATES.

Nominal Horse-power of Boiler.	"Effective" Area of Surface of Boiler in Square Feet.		Area of Fire-grate in Square Feet.	
	Total.	Per Horse-power.	Total.	Per Horse-power.
3	59	19·5	3·7	1·22
4	72	18·C	4·5	1·12
5	85	17·0	5·3	1·06
6	97	16·2	6·0	1·00
7	109	15·8	6·8	·972
8	121	15·1	7·6	·950
9	132	14·7	8·2	·917
10	143	14·3	9·0	·900
12	165	13·8	10·3	·859
14	187	13·4	11·7	·836
15	198	13·2	12·4	·827
16	208	13·0	13·0	·813
18	229	12·7	14·3	·794
20	250	12·5	15·6	·780
22	270	12·2	16·9	·768
24	290	12·1	18·2	·758
25	300	12·0	18·8	·752
26	310	11·9	19·4	·746
28	330	11·8	20·6	·743
30	350	11·6	21·8	·726
35	398	11·4	24·9	·711
40	446	11·1	27·9	·697
45	494	11·0	30·9	·687
50	542	10·8	33·8	·676
55	588	10·7	36·7	·667
60	634	10·6	39·6	·660
(1)	(2)	(3)	(4)	(5)

TABLE 50.—Of the PROPORTIONS of CORNISH BOILERS, with Single-riveted Joints.

Nominal Horse-power.	Body of Boiler. Diameter inside.		Body of Boiler. Thickness of Plate.	Body of Boiler. Safe Internal Pressure in lbs. per Sq. Inch.	Flue, or Fire-tube. No. of Flues.	Flue, or Fire-tube. Diameter inside.		Flue, or Fire-tube. Thickness of Plate.	Flue, or Fire-tube. No. of Rings.	Flue, or Fire-tube. Safe External Pressure in lbs. per Sq. Inch.	Length of Boiler in Feet.	Effective Area of Boiler in Sq. Feet.	Fire-grate. Length.		Fire-grate. Width.		Fire-grate. Area in Sq. Feet.	Area of Flues in Sq. Inches.	Diameter of Safety-valve for 45 lbs. Steam.
	ft.	in.				ft.	in.						ft.	in.	ft.	in.			
3	3	6	5/16	56	1	2	0	1/4	1	67	8	58	1	9	2	0	3·5	340	1½
4	3	6	5/16	56	1	2	0	5/16	1	87	10	73	2	3	2	0	4·5	340	2
5	3	6	5/16	56	1	2	0	5/16	1	73	12	87	2	9	2	0	5·5	340	2
6	4	0	3/8	58	1	2	3	3/8	1	97	12	99	2	9	2	3	6·2	388	2
8	4	6	7/16	61	1	2	6	3/8	1	87	13	120	3	0	2	6	7·5	435	2¼
10	4	6	7/16	61	1	2	6	3/8	1	70	16	148	3	6	2	6	8·8	435	2¼
10	5	0	7/16	55	1	2	9	3/8	1	73	14	143	3	3	2	9	8·9	480	2¼
12	5	0	7/16	55	1	2	9	7/16	1	83	16	163	3	9	2	9	10·3	480	2½
14	5	0	7/16	55	1	2	9	7/16	1	74	18	184	4	3	2	9	11·7	480	2½
16	5	0	7/16	55	1	2	9	7/16	1	67	20	204	4	9	2	9	13·1	480	2½
18	5	6	7/16	50	1	3	0	7/16	1	61	20	224	4	9	3	0	14·3	526	2¾
20	5	6	7/16	50	1	3	0	7/16	1	55	22	246	5	3	3	0	15·7	526	3
22	5	6	7/16	50	1	3	0	3/8	2	54	24	269	5	6	3	0	16·5	526	3¼
22	6	0	1/2	52	2	2	3	3/8	1	61	19	269	3	9	4	6	16·9	665	3¼
24	6	0	1/2	52	2	2	3	3/8	1	56	21	297	4	0	4	6	18·0	665	3½
27	6	0	1/2	52	2	2	3	3/8	2	76	23	325	4	6	4	6	20·2	665	3½
30	6	0	1/2	52	2	2	3	3/8	2	70	25	354	4	9	4	6	21·4	665	3¾
30	6	6	1/2	48	2	2	6	3/8	2	68	23	357	4	3	5	0	21·3	730	3¾
35	6	6	1/2	48	2	2	6	3/8	2	60	26	403	5	0	5	0	25·0	730	4
40	6	6	1/2	48	2	2	6	3/8	2	54	29	450	5	6	5	0	27·5	730	4
40	7	0	9/16	50	2	2	9	7/16	2	74	27	456	5	0	5	6	27·5	794	4
45	7	0	9/16	50	2	2	9	7/16	2	67	30	506	5	6	5	6	30·3	794	4¼
45	7	6	9/16	47	2	3	0	7/16	2	68	27	493	5	0	6	0	30·0	857	4¼
50	7	6	9/16	47	2	3	0	7/16	2	61	30	547	5	6	6	0	33·0	857	4½
55	7	6	9/16	47	2	3	0	7/16	2	55	33	602	6	0	6	0	36·0	857	5
(1)	(2)		(3)	(4)	(5)	(6)		(7)	(8)	(9)	(10)	(11)	(12)		(13)		(14)	(15)	(16)

$3 \cdot 14 \times 7 \cdot 5 \times 30 \div 2 = 352$ square feet of actual surface, and as by (115) the effective surface in such a case is $\frac{3}{4}$ths of the actual, we have $352 \times \cdot 75 = 264$ square feet effective. Then the two tubes have an actual surface of $3 \cdot 14 \times 3 \times 30 \times 2 = 566$ square feet,—equivalent to $566 \div 2 = 283$ square feet effective;—the sum is $264 + 283 = 547$ square feet or $547 \div 50 = 11$ square feet per nominal horse-power. Tables 49 and 50 have been calculated by this rule.

(119.) "*Boilers for Steam-engines.*"—It is very desirable that in all cases the power of a boiler should be estimated by the cubic feet of water evaporated to steam per hour rather than by the horse-power, which is easily done when the diameter of the cylinder, &c., &c., is known. Thus with a 12-inch cylinder, 2 feet stroke, 50 revolutions per minute, cutting off 45 lbs. steam at $\frac{1}{3}$rd; the area of cylinder being $\cdot 7854$ square foot, and the bulk of 45 lbs. steam = 439 for water 1 by Table 71, we shall require $\frac{\cdot 7854 \times 4 \times 50 \times 60}{3 \times 439} = 7 \cdot 15$ cubic feet of water per hour.

(120.) A common high-pressure engine with 40 lbs. steam, working without expansion, except a small amount obtained by lap of the slide, consumes about a cubic foot of water per nominal horse-power. Thus an engine with $12\frac{1}{4}$-inch cylinder, 24 inches stroke, cutting off at 17 inches by lap of slide, with 40 revolutions per minute, and 40 lbs. steam, would show by the indicator about 18 nett horse-power; equal to 12 nominal horse-power by the ratio given in (118). The area of the cylinder being 118 square inches, and the volume of 40 lbs. steam by Table 71 being 476 for water 1, we have $\frac{118 \times 17 \times 2 \times 40 \times 60}{1728 \times 476}$ $= 11 \cdot 98$, say 12 cubic feet of water per hour, being 1 cubic foot per nominal, or $12 \div 18 = \cdot 667$ cubic foot per nett indicated horse-power per hour. The steam was in this case cut off at $17 \div 24 = \cdot 708$, and we may admit $\cdot 7$ as the expansion ordinarily obtainable by lap of slide alone.

(121.) "*Expansive Steam.*"—When steam is cut off before the piston has completed its stroke, the *mean* pressure throughout the stroke is reduced: thus, when 50 lbs. steam is cut off at half

stroke, the pressure during the first half of the stroke is of course 50 lbs.; during the last half the *mean* pressure is reduced by expansion to 30 lbs., hence the mean pressure throughout the whole stroke becomes $(50 + 30) \div 2 = 40$ lbs. above atmosphere.

Let P = total pressure above vacuum, obtained in round numbers by adding 15 to the pressure above atmosphere in pounds per square inch; P′ = back pressure above a vacuum in pounds per square inch; E = expansion or the stroke of the piston divided by the distance it has travelled when steam it cut off; H = hyperbolic logarithm of E, which is given by Table 51; p = mean pressure (above atmosphere) throughout the stroke in pounds per square inch, then

$$p = \frac{(P \times H) + P}{E} - P'.$$

Thus with 50 lb. steam cut off at half stroke, with a back pressure of the atmosphere only, $P = 50 + 15 = 65$; $P' = 15$; $E = 2$; $H = \cdot 693$; and we obtain $p = \frac{(65 \times \cdot 693) + 65}{2} - 15 = 40$ lbs. mean pressure throughout the stroke. Table 52

TABLE 51.—Of HYPERBOLIC LOGARITHMS for EXPANSIVE STEAM.

Expansion, E.	Cut-off.	Hyp. Log., H.	Expansion, E.	Cut-off.	Hyp. Log., H.	Expansion, E.	Cut-off.	Hyp. Log., H.
1·10	..	·095	2·2	..	·787	4·5	..	1·504
1·11	$\frac{9}{10}$	·104	2·25	..	·811	4·75	..	1·558
1·20	..	·182	2·3	..	·832	5·0	$\frac{2}{10}$	1·609
1·25	$\frac{8}{10}$	·223	2·4	..	·874	5·25	..	1·658
1·30	..	·263	2·5	$\frac{4}{10}$	·916	5·5	..	1·705
1·33	$\frac{3}{4}$	·287	2·6	..	·954	5·75	..	1·749
1·4	..	·336	2·7	..	·992	6·0	$\frac{1}{6}$	1·792
1·43	$\frac{7}{10}$	·357	2·75	..	1·012	6·25	..	1·832
1·5	$\frac{2}{3}$	·405	2·8	..	1·028	6·5	..	1·872
1·6	..	·469	2·9	..	1·064	6·75	..	1·909
1·67	$\frac{6}{10}$	·510	3·0	$\frac{1}{3}$	1·098	7·0	$\frac{1}{7}$	1·946
1·7	..	·530	3·25	..	1·118	7·25	..	1·981
1·75	..	·560	3·33	$\frac{3}{10}$	1·203	7·5	..	2·015
1·8	..	·587	3·5	..	1·253	7·75	..	2·048
1·9	..	·641	3·75	..	1·322	8·0	$\frac{1}{8}$	2·079
2·0	$\frac{1}{2}$	·693	4·0	$\frac{1}{4}$	1·386	9·0	$\frac{1}{9}$	2·197
2·1	..	·741	4·25	..	1·447	10·0	$\frac{1}{10}$	2·302

has been calculated by this rule. It should be observed that with non-condensing engines the exhaust steam is sometimes used for heating purposes (230) and extra back-pressure is allowed for convenience, say 5 lbs. per square inch is thus allowed, then P′ becomes $15 + 5 = 20$ lbs., &c. With condensing engines the vacuum is generally more or less imperfect, say it is only 26 inches of mercury, or about 13 lbs. per square inch: then $P' = 15 - 13 = 2$ lbs., &c.

TABLE 52.—Of the MEAN PRESSURE of EXPANSIVE STEAM.

Initial Pressure above Atmosphere in lbs. per Square Inch.	Point of Stroke at which the Steam is cut off.											
	$\frac{1}{10}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{1}{3}$	$\frac{4}{10}$	$\frac{1}{2}$	$\frac{6}{10}$	$\frac{2}{3}$	$\frac{7}{10}$	$\frac{3}{4}$	$\frac{8}{10}$
	Mean Pressure of Steam throughout the Stroke.											
15	−5·1	·66	2·9	4·8	6·0	8·0	10·4	12·0	13·1	13·4	14·0	14·3
20	−3·5	3·27	5·9	8·1	9·5	11·8	14·6	16·7	17·6	18·2	18·8	19·2
25	−1·8	5·88	8·8	11·4	13·0	15·6	18·8	21·2	22·3	22·9	23·6	24·1
30	−0·15	8·5	11·8	14·7	16·5	19·5	23·0	25·8	27·0	27·7	28·4	29·0
35	1·50	11·1	14·8	18·0	20·0	23·3	27·3	30·3	31·6	32·4	33·2	33·9
40	3·15	13·7	17·8	21·3	23·5	27·1	31·5	34·8	36·3	37·2	37·6	38·8
45	4·80	16·3	20·8	24·6	27·0	31·0	35·8	39·4	41·0	41·9	42·9	43·7
50	6·45	19·0	23·8	27·9	30·5	34·8	40·0	43·9	45·6	46·7	47·2	48·6
55	8·10	21·5	26·7	31·2	34·0	38·6	44·2	48·4	50·3	51·4	52·5	53·5
60	9·75	24·1	29·7	34·5	37·5	42·5	48·5	53·0	55·0	56·2	57·4	58·4
65	11·4	26·7	32·7	37·8	41·0	46·3	52·7	57·5	59·6	60·9	62·2	63·2
70	13·0	29·4	35·7	41·1	44·5	50·1	57·0	62·0	64·3	65·7	66·0	68·1
75	14·7	32·0	38·7	44·4	48·0	54·0	61·2	66·5	69·0	70·4	71·9	73·1
80	16·3	34·6	41·7	47·7	51·5	57·8	65·5	71·1	73·6	75·2	76·6	78·0
90	19·6	39·8	47·7	54·3	58·5	95·5	73·9	80·1	83·0	84·6	86·3	87·7
100	23·0	45·0	53·7	60·9	65·5	73·1	82·4	89·2	92·3	94·1	96·0	97·5
120	29·5	55·4	65·5	74·1	79·4	88·5	99·3	107·0	111·5	113·2	115·6	117·0
150	39·5	71·1	83·4	94·0	100·4	111·5	124·7	134·2	139·6	141·6	144·7	146·4

(122.) "*Economy of Expansive Steam.*"—When an engine acts absolutely without expansion, the steam retains its full pressure to the end of the stroke, and escapes into the atmosphere, still containing in itself a large amount of power which is lost. To obtain all the available power out of high-pressure steam it should be cut off at such a part of the stroke, that the terminal pressure is that of the atmosphere. The point of the stroke at which the steam must be cut off to effect that purpose is governed by the pressure of the steam: thus steam cut off at

$\frac{1}{5}$th must have a *total* pressure of 5 atmospheres or $15 \times 5 = 75$ lbs. above a vacuum, or $75 - 15 = 60$ lbs. above the atmosphere. When steam is cut off at $\frac{3}{4}$, $\frac{2}{3}$, $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{4}$, $\frac{1}{5}$, $\frac{1}{7}$, and $\frac{1}{10}$th of the stroke, the pressure of steam should be 5, $7\frac{1}{2}$, 15, 30, 45, 60, 90, and 135 lbs. per square inch above atmosphere respectively: the terminal pressure will then in all cases be that of the atmosphere. If the steam is cut off later the terminal pressure will be above the atmosphere: thus 60 lbs. steam, having a total pressure of $60 + 15 = 75$ lbs., cut off at $\frac{1}{3}$rd of the stroke would have a terminal pressure of $75 \div 3 = 25$ lbs. above vacuum, or $25 - 15 = 10$ lbs. above atmosphere, and escaping at that pressure there would be a certain loss of effect: see col. 4 of Table 53. If, on the other hand, expansion be carried too far, the terminal pressure is reduced below the atmosphere, resulting in a loss of useful effect and causing trouble with the slide valve by back pressure; thus 30 lbs. steam or 45 lbs. above vacuum cut off at $\frac{1}{5}$th, would give a terminal pressure of $45 \div 5 = 9$ lbs. above vacuum, or $15 - 9 = 6$ lbs. *below* atmosphere. It is impossible in practice to maintain the steam exactly at the uniform pressure required for the most perfect economy, and it is expedient in most cases to allow a higher pressure than that due to the expansion.

Table 53 gives the relative economy of steam in high-pressure expansive engines with varying rates of expansion; col. 2 gives the ratio of the power of the same engine as governed by the grade of expansion. Thus with 45 lbs. steam, an engine which gave 10 horse-power when the steam acted without expansion, would give $9 \cdot 31$ horse-power when the steam was cut off at $\frac{7}{10}$ths, and $5 \cdot 42$ horse-power when cut off at a $\frac{1}{4}$th; the relative cost of a unit of power is $1 \cdot 00$; $\cdot 7 \div \cdot 931 = \cdot 752$; and $\cdot 25 \div \cdot 462 = \cdot 542$ respectively, as in col. 3.

(123.) We found in (120) that an engine cutting off 40 lbs. steam at $\cdot 7$ required 1 cubic foot of water per nominal horse-power: with full steam throughout the stroke we should have $1 \div \cdot 7 = 1 \cdot 4286$ cubic foot, and the power being by Table 52 increased in the ratio of 40 to $37 \cdot 2$ we have $1 \cdot 4286 \times 37 \cdot 2 \div 40 = 1 \cdot 328$ cubic foot of water per nominal horse-power. The cols. 5 and 6 in Table 53 give the cubic feet of water evaporated per nominal and indicated horse-power with various rates of

expansion; they also show that the economy of steam in non-condensing engines increases with the pressure when the rate of expansion is duly proportioned thereto. Thus with 30, 45, 60, and 75 lbs. steam expanded to a terminal pressure of 0 in all cases, we require by col. 5, ·88, ·69, ·593, and ·528 cubic foot of water per nominal horse-power respectively.

Say we have an engine with 35 lbs. steam, cut off at $\frac{7}{10}$ths by lap of slide, and doing a certain amount of work, and we desire

TABLE 53.—Of the ECONOMY of EXPANSIVE STEAM in NON-CONDENSING STEAM-ENGINES.

Steam cut off at	Pressure 30 lbs. per Square Inch.					Pressure 45 lbs. per Square Inch.				
	Ratio of Power.	Cost of Unit of Power.	Pressure at end of Stroke.	Cubic Feet of Water per Horse-power. Nominal.	Cubic Feet of Water per Horse-power. Indicated.	Ratio of Power.	Cost of Unit of Power.	Pressure at end of Stroke.	Cubic Feet of Water per Horse-power. Nominal.	Cubic Feet of Water per Horse-power. Indicated.
Full	1·000	1·000	30	1·46	·97	1·000	1·000	45	1·28	·85
$\frac{8}{10}$	·967	·827	21	1·21	·81	·971	·824	33	1·05	·70
$\frac{3}{4}$	·947	·792	18·7	1·16	·77	·953	·787	30	1·00	·67
$\frac{7}{10}$	·923	·758	16·5	1·11	·74	·931	·752	27	·96	·64
$\frac{5}{8}$	·900	·741	15	1·08	·72	·913	·730	25	·93	·62
$\frac{6}{10}$	·860	·698	12	1·02	·68	·875	·686	21	·88	·59
$\frac{1}{2}$	·767	·652	7·5	·95	·63	·795	·629	15	·80	·533
$\frac{4}{10}$	·650	·615	3	·90	·60	·690	·580	9	·74	·493
$\frac{1}{3}$	·550	·606	0	·88	·58	·600	·555	5	·71	·473
$\frac{3}{10}$	..	..	..	..	..	·547	·549	3	·70	·467
$\frac{1}{4}$	..	..	..	..	..	·462	·542	0	·69	·460

Steam cut off at	Pressure 60 lbs. per Square Inch.					Pressure 75 lbs. per Square Inch.				
Full	1·000	1·000	60	1·19	·793	1·000	1·000	75	1·13	·753
$\frac{8}{10}$	·973	·822	45	·978	·652	·975	·821	57	·928	·619
$\frac{3}{4}$	·957	·784	41·2	·933	·622	·959	·782	52·5	·884	·589
$\frac{7}{10}$	·937	·747	37·5	·889	·593	·939	·746	48	·843	·562
$\frac{5}{8}$	·917	·727	35	·865	·577	·920	·725	45	·819	·546
$\frac{6}{10}$	·883	·679	30	·808	·539	·887	·676	39	·764	·509
$\frac{1}{2}$	·808	·619	22·5	·737	·491	·816	·613	30	·693	·462
$\frac{4}{10}$	·708	·565	15	·672	·448	·720	·556	21	·628	·419
$\frac{1}{3}$	·625	·533	10	·634	·423	·640	·521	15	·589	·393
$\frac{3}{10}$	·575	·522	7·5	·621	·414	·592	·507	12	·573	·382
$\frac{1}{4}$	·495	·505	3·7	·601	·401	·516	·488	7·5	·551	·367
$\frac{1}{5}$	·402	·498	0	·593	·395	·427	·468	3	·529	·353
$\frac{1}{6}$	..	..	..	..	..	·357	·467	0	528	·352
(1)	(2)	(3)	(4)	(5)	(6)	(2)	(3)	(4)	(5)	(6)

to economize fuel by using steam of a higher pressure, say 75 lbs. acting expansively by an expansion-slide, &c. By Table 52, 35 lbs. steam cut off at ·7 gave 32·4 lbs. mean pressure throughout the stroke; we require this same mean pressure with 75 lbs. steam in order to do the same work, and Table 52 shows that it must be cut off at $\frac{1}{5}$th or ·2, the mean pressure being then 32·0 lbs. Now, by Table 71, taking the capacity of the cylinder for the sake of illustration at 1 cubic foot, we should have with 35 lbs. steam ·7 ÷ 521 = ·001344 cubic foot of water evaporated to steam, and with 75 lbs. steam ·2 ÷ 299 = ·00067 cubic foot, or just one-half: so that we should do the same work as before with half the fuel.

(124.) With pressures above atmosphere of 10, 15, 20, 30, 40, 45, 50, 60, and 75 lbs. acting without expansion, the relative cost of a "unit" of power would be 1·0, ·794, ·688, ·583, ·529, ·510, ·495, ·473, and ·450 respectively. When the steam is cut off at such a point as to give a terminal pressure of 0 in all cases, the economy increases with the pressure of the steam in a still higher ratio: with the same pressure as before the relative cost of a unit of power becomes 1·0, ·726, ·588, ·450, ·376, ·350, ·330, ·288, and ·268 respectively, showing that 50 lbs. steam is more economical than 10 lbs. steam in the ratio of ·33 to 1 or $\frac{1}{3}$rd, and 75 lbs. steam about $\frac{1}{4}$th. In practice the ratio would be even higher than this, for we have allowed nothing for friction of engine, &c., which would be much greater in proportion to the power with the low pressures.

The areas of the cylinders of similar non-condensing engines, differing only in the rate of expansion, must vary in the ratio given by col. 2 of Table 53; thus an engine cutting off 75 lbs. steam at $\frac{1}{5}$th must have a cylinder of larger *area* than a similar engine cutting off the same steam at $\frac{7}{10}$ths and doing the same work, in the ratio of 939 to 427.

Table 52 may be applied to condensing engines by adding the pressure due to the vacuum which is constant throughout the stroke, to the mean pressure due to the expanding steam given by that table. Thus, say we have a vacuum of 28 inches of mercury or 14 lbs. nearly per square inch, and 25 lbs. steam cut off at $\frac{1}{3}$rd. The mean pressure of the steam by Table 52

is 13 lbs. per square inch, which added to that due to the vacuum gives a total of $13 + 14 = 27$ lbs. mean pressure throughout the stroke. By the rule in (121), H being $1 \cdot 098$ by Table 51, $P = 25 + 15 = 40$, $P' = 15 - 14 = 1$, and $E = 3$, p becomes $\frac{(40 \times 1 \cdot 908) + 40}{3} - 1 = 27$ lbs. as before.

Having thus shown how variable and indefinite a term "horse-power" is as applied to boilers, we may repeat with emphasis the recommendation in (119) to calculate their power by the water evaporated, whenever it is possible to do so.

(125.) "*Feed-water Heater.*"—The economy of fuel for a steam-boiler depends in part on the temperature at which the feed-water is supplied to it. The water heater to a non-condensing engine generally consists of a water jacket on the exhaust pipe, and when the exhaust steam escapes at the pressure of the atmosphere, its temperature will be 212°, which of course is the maximum to which the feed-water can be raised. But by placing the heater in the flue leading from the boiler to the chimney, where the air is about 500° or more (100), a much higher temperature may be obtained; there is, however, considerable practical difficulty in carrying out that arrangement.

The mean temperature of water in this climate is about 50°, as shown by Table 34, and to convert a pound of water at 50° to steam (18) requires $1178 - 50 = 1128$ units of heat. With a water heater, raising the feed-water to 212°, we require $1178 - 212 = 966$ units, or $966 \div 1128 = \cdot 86$, or 86 per cent., showing a saving of 14 per cent. by the use of such a heater. But with 50 lbs. steam, the temperature by Table 71 is 298°, and with a heater in the flue, raising the feed-water to that temperature, we require $1178 - 298 = 880$ units, or $880 \div 1128 = \cdot 78$, or 78 per cent., showing a saving of 22 per cent.

With a condensing engine, the feed-water is commonly taken from the hot-well at about 100°; here we require $1178 - 100 = 1078$ units, or $1078 \div 1128 = \cdot 955$, or $95 \cdot 5$ per cent., showing a gain of $4 \cdot 5$ per cent. over cold water, but a loss of $1078 \div 966 = 1 \cdot 116$, or $11 \cdot 6$ per cent., as compared with a high-pressure boiler fed with water at 212°, losing thus, much of the gain from condensation (124).

(126.) "*Superheated Steam.*"—The object in superheating steam is to prevent the formation of water by condensation in the pipes and cylinder of a steam-engine, which not only causes a loss of fuel, but is obstructive to effective working. Taking the case in (321) of a 4-inch pipe 100 feet long, with 35 lbs. steam, and admitting the loss by the cylinder of the engine to be equivalent to 40 feet more, we have $587 \times 140 = 82180$ units per hour, which is equal to the condensation of $82180 \div 966 = 85$ lbs. of water. By placing a superheater at the end of the pipe next to the boiler, this condensation may be wholly avoided, and we can easily calculate the temperature to which the steam must be heated to effect that purpose. It is very desirable that it should not be heated very much higher than is necessary, as an extreme temperature would be destructive to the lubricating oil and packing of glands, &c.

Say that in our case the pipe passes 50 horse-power of steam, or 50 cubic feet of water evaporated to steam, per hour: then we have $62 \cdot 32 \times 50 = 3116$ lbs. of steam per hour, and the specific heat of steam with constant pressure being $\cdot 475$ by Table 5, and as it has to receive 82,180 units of extra heat, being the amount lost by the pipe and cylinder, &c., its temperature must be raised $\frac{82180}{3116 \times \cdot 475} = 55°$. Thus the 35 lbs. steam leaves the boiler at its normal temperature of 280°, is raised by the superheater to $280 + 55 = 335°$, and passing through the pipe, &c., is cooled down again to 280°, the heat thus parted with supplying the loss by the surface of the pipe, and preventing the condensation which would otherwise have occurred.

If the superheater is placed in the boiler flue, the heated air will be cooled considerably by it. Admitting 10 lbs. of coal per horse-power, and $22 \cdot 83$ lbs. of air per pound of coal (98), we have in our case $22 \cdot 83 \times 50 \times 10$ $11415 =$ lbs. of air in the flue per hour; and as this has to yield 82,180 units of heat to the steam, and the specific heat of air being $\cdot 238$, it would be cooled $\frac{82180}{11415 \times \cdot 238} = 30°$, or to $550 - 30 = 520°$. The mean temperature of the heated air as it passes through the

superheater is $(550 + 520) \div 2 = 535°$, and that of the steam being $(280 + 335) \div 2 = 308°$; the difference is $535 - 308 = 227°$, the ratio for which by Table 105 is $1 \cdot 7$, and the value of A for say 3-inch pipe by Table 99 being $\cdot 6256$, we have $\cdot 6256 \times 1 \cdot 7 \times 227 = 241$ units per square foot per hour. Hence we require $82180 \div 241 = 341$ square feet of tube surface; and the outside (319) area of a 3-inch pipe being say $\cdot 9$ of a square foot, we require $341 \div \cdot 9 = 380$ feet run of 3-inch pipes; see (282), (315).

Admitting 8190 units as the *useful* effect of a pound of coal, as per Table 47, the saving of fuel by superheating is $82180 \div 8190 = 10$ lbs. of coal per hour, or about 1 horse-power, and what is perhaps of more importance, the formation of obstructive water by condensation is avoided. With the proportions we have given, the steam enters the cylinder of the engine at the same temperature as it left the boiler, and objectionable overheating, which is the great drawback to superheating, is prevented.

(127.) "*Furnaces to Steam-boilers.*"—The consumption of fuel per square foot of grate may be varied very considerably when the draught is good, without any sensible effect on the economy: in ordinary cases 12 or 14 lbs. of coal per square foot per hour is a good average quantity. A *very* large grate and thin fire is very objectionable, for without great care parts of the grate are liable to become uncovered, whereby a large volume of air passes through the furnace unconsumed, resulting in a great loss of useful effect (108).

The same causes which render the heating surface of boilers, per horse-power, a variable quantity, affect the area of fire-grate also; in fact, the area of grate should be proportional to the area of the boiler, say $\frac{1}{18}$th of the *effective* area. We have therefore the rule:—

$$G = \left(H + (\sqrt{H} \times 2 \cdot 5\right) \div 2,$$

in which H = the nominal horse-power of the boiler, or cubic feet of water at 60° evaporated to steam per hour; and G = the area of grate in square feet. Col. 4 of Table 49 has been

calculated by this rule, also col. 14 of Table 50. Thus for the 50-horse boiler in (118) the proper area of fire-grate would be $\left(50 + (\sqrt{50} \times 2 \cdot 5\right) \div 2 = 33 \cdot 8$ square feet, say $5 \cdot 5 \times 6 = 33$ square feet. Allowing 13 lbs. of coals per square foot, we have $33 \times 13 = 430$ lbs. of coals per hour, or $430 \div 50 = 8 \cdot 6$ lbs. per cubic foot of water, or per nominal horse-power (118). Col. 3 of Table 43 gives 8·62 lbs. of Newcastle coal per cubic foot of water evaporated from 60° to steam.

(128.) "*Fire-bars.*"—Fire-bars should be short, thin, and deep; the length in most cases should not exceed 3 feet; a long fire-bar is apt to be distorted by the heat, and give trouble, and this will not be obviated by increasing the thickness. A thin bar will stand better than a thick one, despite its apparent weakness; but the fact is, that a fire-bar is cooled by the passage of cold air on both sides of it, and thus a thin bar is cooled more effectively than a thick one. Fig. 12 gives good general proportions for fire-bars; the dimensions apply to all lengths except the depth at the centre, which is given by Table 54,

TABLE 54.—Of the PROPORTIONS of FIRE-BARS for STEAM-BOILERS, &c., &c.

Length.		Depth.	Weight.	Weight per Square Foot.	Length.		Depth.	Weight.	Weight per Square Foot.
ft.	in.	inches.	lbs.	lbs.	ft.	in.	inches.	lbs.	lbs.
1	0	2¼	6·4	64	3	3	4¾	29	89
1	3	2¾	8·1	64	3	6	5	32	91
1	6	3	9·5	64	3	9	5¼	35	93
1	9	3¼	11·4	65	4	0	5½	39	98
2	0	3½	*13	65	4	3	5¾	42	100
2	3	3¾	*15½	69	4	6	6	47	104
2	6	4	*18	72	4	9	6¼	52	109
2	9	4¼	*20¾	76	5	0	6½	*56¾	113·5
3	0	4½	*23¾	79					

which is calculated by the rule $L + 1 \cdot 5 = d$, in which L = the length of bar in feet, and d the depth at the centre in

* Actual weights, obtained by weighing the castings.

inches. The fourth column shows that short bars weigh less per *square* foot than long ones.

"*Dead-plate, &c.*"—The object of the dead-plate is principally to keep the fire away from the furnace-front, and prevent it becoming unduly heated. Its width may vary with the size of the boiler, as is shown by Figs. 36 to 43, being 6 inches wide in small, 9 inches in medium, and 12 inches in large boilers. For the same purpose, a screen-plate as at D in Fig. 8 is a useful addition to the furnace-door. The figures give the longitudinal section of furnaces. The bearing-bars should not be firmly fixed to the boiler, as at A, Fig. 20, as is frequently done, for in that case the expansion by heat is very apt to work the screws loose, and cause leakage; but should rest loosely in a pocket formed of angle-iron, as at B. The back-bridge may be of thick cast iron, or of fire-brick, as in the figures.

(129.) "*Steam-chest.*"—The primary object of a steam-chest is to form a steam reservoir, where the steam may be quietly separated from the water, which otherwise is very apt to *prime* over with it (72). In single-flued Cornish boilers, the steam-space is very small, and a steam-chest is essential; but in the double-flued boilers, as in Fig. 36, &c., there is abundant steam-room without a steam-chest: nevertheless it is advisable to adopt it in all cases; for with a little management, the man-hole and the steam-valves, &c., may all be fixed upon it, as in Fig. 21, and in that case no openings whatever are required in the body of the boiler. The whole may be floored over with York paving, and great neatness obtained. This cannot, however, be done very well with less diameter than 2 ft. 6 in. The top of the chest will require for high-pressure steam a bar of T iron across to strengthen it, a second man-hole must be made in the body of the boiler, immediately beneath the one in the top of the chest; but in no case must the whole of the metal in the body covered by the chest be cut out, as the boiler would thereby be seriously weakened.

(130.) "*Safety-valves.*"—The velocity with which steam issues from a boiler, &c., is the same as that of a body falling by gravity from the height of a homogeneous column of steam, having throughout the same density as at the orifice. Say we take the

case of steam 40 lbs. per square inch above the atmosphere. By Table 38, 1 lb. pressure is equal to 2·3 feet of water, therefore the pressure in our case is $2 \cdot 3 \times 40 = 92$ feet of water. By Table 71, steam at 40 lbs. is 476 times the bulk of water; the height of the column of steam is therefore $92 \times 476 = 43792$ feet, and the velocity due to that height by the rule for falling bodies ($\sqrt{H} \times 8$) is, in our case, $\sqrt{43792} \times 8 = 1672$ feet per second; but when the orifice is made in a thin plate, the issuing jet of steam suffers a contraction, so that its area is reduced, according to the experiments of Daubuisson, to ·65, the actual area being 1·0; the discharge by an aperture 1 inch square, per hour, is therefore $1672 \times 3600 \times \cdot 65 \div 144 = 27170$ cubic feet of 40 lbs. steam,—equal to $27170 \div 476 = 57 \cdot 08$ cubic feet of water, or 57 horse-power according to (118).

(131.) An experiment was made with a double-flued Cornish boiler 24 ft. 3 in. long, 6 ft. 6 in. diameter, with two 2 ft. 6 in. flues, and 30 feet of fire-grate (about 35 horse-power, by our Table 50). The steam was discharged by an aperture 1 inch square in a thin plate, and by *forced* firing 57·5 cubic feet of water were evaporated in an hour—the pressure of steam varying from 35 to 43 lbs., agreeing remarkably with the preceding calculation. A $3\frac{3}{4}$-inch safety-valve on the same boiler, loaded to 30 lbs., discharged the same quantity of steam at 46 lbs. pressure. The valve whose angle was 45° was raised vertically ·122 inch, as in Fig. 22; the width of the annular discharging-orifice was therefore $\cdot 122 \div 1 \cdot 414 = \cdot 0863$: and the circumference of $3\frac{3}{4}$ being 11·78, we have $\cdot 0863 \times 11 \cdot 78 = 1 \cdot 016$ square inch as the area, or very nearly the same as before. It will be observed that although the valve was loaded to 30 lbs., and would doubtless begin to blow off at that pressure, yet the pressure increased to 46 lbs. before the valve was able to carry off the steam, giving an increase of more than 50 per cent. This great increase arises from the fact that the valve was too small for the quantity of steam. It was large enough for the boiler when working at its nominal power of 35 horses, but not for 57·5 horse-power to which it was forced, for which, with 40 lbs. steam, we require by Table 55 a 5-inch safety-valve. Even

with that size, the pressure would increase considerably (perhaps 15 or 20 per cent.) above that to which the valve was loaded. Nor can this be obviated without using valves of enormous and impracticable size.

(132.) It will be seen from this that there is no precise standard for the size of a safety-valve: all that can be done is to fix upon a size that will not suffer the pressure to rise to a dangerous extent. The area of a safety-valve should be proportional to the area of fire surface in the boiler, and should be determined by that rather than by the horse-power of the boiler. We may admit as the result of experience that a 20-horse boiler (as in Table 50) with 45 lbs. steam requires one valve 3 inches diameter. Now that boiler has an area of 246 square feet, or $246 \div 3^2 = 27$ square feet per circular inch of valve, or 3 square yards; and for this pressure we may take the rule $A = d^2 \times 27$, in which d = the diameter of the valve in inches, and A = the *effective* area of the boiler in square feet. The fourth column in Table 55 has been calculated by this rule.

(133.) We can calculate from this the areas with other pressures, say with the same boiler, we take 7 lbs. steam. We find by Table 71 that we have to discharge a larger volume of steam than with 45 lbs. pressure, in the ratio of 1138 to 439, so that the 3-inch valve, which was equal to 246 square feet of boiler surface with 45 lbs. steam, would now be equal to only $246 \times 439 \div 1138 = 95$ square feet, if the velocity of discharge was the same, which it is not, for (130) with 45 lbs. steam the velocity is $\sqrt{45 \times 2 \cdot 3 \times 439 \times 8} = 1704$ feet per second, but with 7 lbs. steam, $\sqrt{7 \times 2 \cdot 3 \times 1138 \times 8} = 1090$ feet per second, and the area of boiler surface for a 3-inch valve is thus reduced to $95 \times 1090 \div 1704 = 60 \cdot 7$ square feet, or about one-fourth of the area with 45 lbs. steam. The different columns in the first part of Table 55 have been calculated in this way, and they give the areas of boiler surface due to standard sizes of safety-valves.

(134.) By Table 49 we may easily find the diameter of safety-valve from those areas: thus for 20 horse-power that table gives 250 square feet, the nearest number to which in Table 55 is 243, which is equal to a 3-inch for 45 lbs. steam, 4-inch for 25,

TABLE 55.—Of the SIZES of SAFETY-VALVES for STEAM-BOILERS.

Diameter of Valve.	Pressure of Steam in Pounds per Square Inch above the Atmosphere.				
	7	25	45	65	100
	"*Effective*" Area of Boiler Surface in Square Feet.				
in.					
1	7	17	27	38	54
1½	15	39	61	84	123
2	27	67	108	149	218
2½	42	105	170	235	343
3	61	151	243	335	491
3½	83	205	331	457	668
4	108	268	432	596	873
4½	137	329	547	755	1105
5	169	418	675	932	1363
5½	203	505	814	1123	1644
6	243	603	972	1341	1963
	Nominal Horse-power of Boiler.				
1	..	..	..	..	3
1½	..	..	3	5	8
2	..	3½	7	10	17
2½	2	7	13	18	29
3	3	11	20	28	45
3½	5	16	28	41	63
4	7	22	39	56	..
4½	9	28	50	..	..
5	12	37	64	..	..
5½	16	46	..	..	..
6	20	57	..	..	..

NOTE.—The *effective* area meant in this table is explained in (113–116); see also Table 49 for the connection between the effective area and the horse-powers of boilers, &c.

and 6-inch for 7 lbs. steam, &c. The second part of Table 55 gives the horse-power of safety-valves of different diameters obtained in this way. Table 50 also gives in col. 16 the proper diameter of safety-valve for different sizes of boilers.

(135.) The forms of safety-valves are variable; they are commonly made conical in both valve and seat as in Fig. 22, but it is much preferable to make the seat very narrow, say $\frac{1}{16}$th of an inch wide, as in Fig. 26. With a wide seat

there is considerable uncertainty in estimating the acting or effective diameter, which is usually intermediate between the large and small diameters of the cone; with a narrow seat this is avoided, and if well executed it is more easy to grind true and keep steam-tight. Safety-valves are frequently made close-topped with a waste-pipe to the chimney; but this is objectionable, because the valve may be leaky, or by careless firing the steam may frequently be allowed to get up too high and blow off without attracting attention. The open-topped form is preferable, and with ordinary care in firing should seldom be found blowing off.

A convenient and inexpensive form of safety-valve is shown by Fig. 25; the valve is spherical, fitted to a cylindrical seat. The line *e, f* is drawn from the point where the sphere and the cylinder touch, at an angle of 45°, and cuts the axis B, E at a point which is the centre of the sphere of which the valve is a part. It follows from this, that the radius of the spherical valve is 1·414 times the radius of the cylindrical seat, so that for valves 1, 2, 3, 4, 5, and 6 inches diameter, the radius of the sphere is ·71, 1·41, 2·12, 2·83, 3·54, and 4·25 inches respectively. The valve might be a simple cone as at F, but the spherical form is the best; it is shown enlarged at H.

In adjusting the weight G, allowance should be made for the weight of the lever and valve alone, which may be done by the application of a Salter's balance at B. Say we had a 3-inch valve for 45 lbs. steam, and that the *effective* weight at B was 12 lbs., and we had to determine the weight at C, the distances A B and A C being $3\frac{1}{4}$ and $19\frac{1}{2}$ inches respectively, or 1 to 6. The area of 3 inches being 7·06, we have $\{7 \cdot 06 \times 45) - 12\} \times 3 \cdot 25 \div 19 \cdot 5 = 51$ lbs. at C.

(136.) "*Dampers.*"—The area of a damper depends on the height of the chimney, and where there is only one boiler it may have the same area as the chimney, if that is properly proportioned to the power of the boiler. Thus for a chimney, as per Table 63, 40 feet high, 12 inches square, we have $144 \div 8 \cdot 1 = 17 \cdot 7$ inches per horse-power, but for a chimney 150 feet high, and 2 feet 6 inches square, $900 \div 108 = 8 \cdot 4$ inches per horse-power only. As an approximate rule, we may give $110 \div \sqrt{H} = A$, in which H = the height of the chimney

in feet, and A = the area of the damper in square inches per horse-power; thus for a chimney 100 feet high we have 110 ÷ 10 = 11 inches per horse-power. The form of damper is arbitrary, and must often be varied to suit the form of the flue, but for ordinary cases we may adopt standard sizes, a convenient proportion being 3 to 1, and thus we have the sizes and powers given in Table 56. The powers of other sizes may be easily calculated by the numbers in the fourth line of the table; thus, say we required the size for a large damper to a set of boilers 300 horse-power for a chimney 100 feet high; the table gives 11 inches per horse-power, and we have 300 × 11 = 3300 square inches for the area required, and if the height was fixed at 6 feet, or 72 inches, the width must be 3300 ÷ 72 = 46 inches, &c., &c.

TABLE 56.—Of the SIZES of DAMPERS to STEAM-BOILERS, with different Heights of Chimney.

Size of the Damper in Inches.	Height of Chimney in Feet.					
	40	60	80	100	120	150
	Square Inches of Damper per Horse-power.					
	17·4	14·2	12·4	11·0	10·0	9·0
	Horse-power of the Boiler.					
6 × 18	6·2	7·6	8·7	9·9	10·8	12
7 × 21	8·5	10	12	13	15	16
8 × 24	11	13	16	18	19	22
9 × 27	14	17	20	22	24	27
10 × 30	17	21	25	28	30	34
12 × 36	25	31	35	40	43	48
14 × 42	34	41	47	53	59	65
16 × 48	44	54	62	70	77	85

(137.) "*Area and Arrangement of Flues.*"—The volume of heated air which has to pass along a boiler flue is proportional to the horse-power, and as in order to give out a given amount of heat, or to be cooled to a fixed temperature, it must be in contact with the boiler a certain *time*, it follows that the velocity of the current should be proportional to the length of the boiler,

so that a particle of air traverses it in the same time, whether the boiler be long or short, and departs into the chimney cooled down to the same temperature in all cases. Let us take three boilers, say 10, 20, and 30 feet long, and 4, 16, and 50 horse-power, as per Table 50; the velocities should therefore be in the ratios 1, 2, 3, and the volumes of air in the ratios 4, 16, 50, or 1, 4, 12 nearly; the areas must therefore be in the ratio of $\frac{\text{Volume}}{\text{Velocity}}$ or $\frac{1}{1} = 1$, $\frac{4}{2} = 2$, and $\frac{12}{3} = 4$, so that the 50-horse boiler requires a flue only four times the area of a 4-horse one with the length of boilers we have taken. If the volume of air is proportional to the horse-power, and the velocity proportional to the length, the area of flue would be in the ratio of Horse-power $\div$ Length.

(138.) But we have seen (117) that the volume of air is not simply proportional to the horse-power, small boilers consuming more fuel per horse-power than large ones, and requiring more air. The area of flue must therefore be made proportional to the area of the fire-grate, or (what is the same thing) to the effective area of the boiler, and the rule becomes

$$a = A \times 47 \div L,$$

in which A = the *effective* area of the boiler in square feet as in (113), L = the length of the boiler in feet, and a = the cross-sectional area of flue in square inches. Thus in our cases, the 4-horse boiler having by col. 11 of Table 50 an area of 73 square feet, requires with a length of 10 feet, $73 \times 47 \div 10 = 343$ square inches of flue, or 86 inches per horse-power; the 16-horse, $204 \times 47 \div 20 = 480$ square inches, or 30 inches per horse-power; and the 50-horse, $547 \times 47 \div 30 = 857$ square inches, or 17·1 inches per horse-power. Col. 15 of Table 50 has been calculated by this rule.

It is remarkable, that with boilers whose power is in the ratio of 4 to 50, or 1 to 12·5, the area of flues is in the ratio 343 to 857, or 1 to 2·5 only; but these proportions will be found to agree with the practice of our best engineers, who in most cases have had only experience to guide them. Thus with a "wheel" draught, the 4-horse flue would be 5 inches at the

top, and 11 inches at the bottom, the mean = 8, and the height measured on the curve being 43 inches, the area is $43 \times 8 = 344$ square inches. With the 50-horse, the flue would be 6 and 14, the mean 10, and the height 85 inches, and we have $10 \times 85 = 850$ square inches. We have in both cases supposed the flue to be carried up to the level of the top of fire-tube, which may generally be admitted with advantage in Cornish boilers.

(139.) "*Modes of Setting Boilers.*"—There are three principal ways in which the flues of a Cornish boiler may be arranged. Fig. 27 shows the best plan: the fire proceeding along the tube to the back of the boiler descends and returns beneath the body to the front, where it splits and passes on both sides to the chimney. This mode of setting is preferable to any other, because the *bottom* of the boiler is more effectively heated, and thereby a better circulation of the water is effected; it requires, however, rather more length in the house than other modes. Fig. 28 is another arrangement, in which the fire splits at the back, returning on both sides to the front, where it descends, and proceeds by one flue to the chimney. Fig. 29 shows a "wheel" draught, and is the most common form of any. The fire returns to the front on one side, passes under the boiler by an opening in the dividing-wall, and passes to the chimney on the other side. There is a practical objection to this plan; if a slight leakage occurs in the body of the boiler, the water will trickle down and saturate the middle wall beneath the bottom, and corrosion will proceed insidiously to a damaging and perhaps dangerous extent.

The flues which receive the first part of the heat should be lined with fire-brick; the boiler should be fixed at such a level that the fire-bars are about 2 feet 6 inches above the stoke-hole floor.

(140.) "*Strength of Steam-boilers.*"—For much of our knowledge of the strength of steam-boilers we are indebted to the venerable Mr. Fairbairn, who gives 34,000 lbs. per square inch as the reduced strain on the solid part of the plate when the joint is breaking through the line of rivet-holes, and as he allows the working strain to be one-sixth of the ultimate strength, we have $34000 \div 6 = 5670$ lbs. as the safe strain.

These data have been commonly taken as standards in calculating the strength of boilers, but Staffordshire plates, which are more commonly used than any other, have considerably less strength than those from which these data were derived. The mean strength of the solid plate in Staffordshire iron is 20 tons, or 44,800 lbs. per square inch; but Mr. Fairbairn's experiments have shown that the metal is damaged by punching, so that the strength of the unpunched plate per square inch being 1·0, that of the metal between the holes is ·7615 in single-riveted joints, and ·933 in double-riveted ones. Much of this reduced strength is lost by the metal being punched out for the rivets; thus with $\frac{3}{8}$ plate, $\frac{11}{16}$ rivets, 2 inches pitch, the space between the rivet-holes is $1\frac{5}{16}$, and the ratio of the area through the line of rivet-holes is to that of the solid plate as $1\frac{5}{16} \div 2$ or $21 \div 32 = \cdot 656$ to 1. Hence the reduced strain on the solid plate when the joint is breaking at the rivet-holes, is with single-riveted joints $44800 \times \cdot 7615 \times \cdot 656 = 22380$ lbs. or 10 tons per square inch, and with double-riveted joints $44800 \times \cdot 933 \times \cdot 656 = 27420$ lbs.

(141.) Taking the working strain at one-sixth of the ultimate strength, we have for single-riveted joints $22400 \div 6 = 3733$ lbs. per square inch, and the rules:—

$$p = 7466 \times t \div d; \qquad t = p \times d \div 7466,$$

and with double-riveted joints, $27420 \div 6 = 4570$ lbs., and the rules:—

$$p = 9140 \times t \div d; \qquad t = p \times d \div 9140,$$

in which p = the safe working pressure in pounds per square inch above atmosphere, t = thickness of plate in inches, and d = inside diameter of boiler in inches. Table 57 has been calculated by these rules, also col. 4 of Table 50; see (145).

In applying these rules for very low pressures, it will be found that the thickness comes out too light to satisfy practical considerations, although sufficient to resist the pressure. Thus for a boiler $6\frac{1}{2}$ feet or 78 inches diameter and 6 lbs. pressure, the thickness by the rule is $6 \times 78 \div 7466 = \cdot 0626$, or $\frac{1}{16}$th inch only, which obviously is excessively too light; in fact, if it were possible to construct the boiler with that thickness, it would not

be able to sustain its own weight and that of the water contained by it.

The rules have therefore certain limitations: in the first place we should not usually make use of plates less than $\frac{3}{16}$ inch thick for steam-boiler work whatever the pressure or diameter; and secondly, with thicknesses of $\frac{3}{16}$, $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, and $\frac{7}{16}$, the maximum diameters should not exceed 4, 5, 6, 7, and 8 feet respectively, the corresponding pressures by the rule being 29, 31, 32, 33, and 34 lbs. per square inch, as per Table 57, which is carried out in accordance with these limitations.

TABLE 57.—Of the STRENGTH of CYLINDRICAL BOILERS made of STAFFORDSHIRE PLATES, with Riveted Joints to resist an Internal or Bursting Pressure.

Thickness of Plate.	Boilers with Single-riveted Joints. Inside Diameter of Boiler in Feet.													
	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	$5\frac{1}{2}$	6	$6\frac{1}{2}$	7	$7\frac{1}{2}$	8
	Working Pressure of Steam in Pounds per Square Inch.													
$\frac{3}{16}$	78	58	47	39	33	29	..	..	..	..	..	..	..	..
$\frac{1}{4}$	104	78	62	52	45	39	35	31	..	..	..	..	..	..
$\frac{5}{16}$	130	97	78	65	56	49	43	39	35	32	..	..	..	..
$\frac{3}{8}$	..	117	93	78	67	58	52	47	42	39	36	33	..	..
$\frac{7}{16}$	..	..	109	91	78	68	61	55	50	45	42	39	36	34
$\frac{1}{2}$	..	..	..	104	89	78	69	62	57	52	48	44	42	39
$\frac{9}{16}$	..	..	..	..	100	88	78	70	64	58	54	50	47	44
	Boilers with Double-riveted Joints.													
$\frac{3}{8}$	191	143	115	95	82	72	64	57	52	48	44	41	..	..
$\frac{7}{16}$	..	167	133	111	95	83	74	67	61	56	51	48	45	42
$\frac{1}{2}$	..	..	152	127	109	95	85	76	69	64	59	55	51	48
$\frac{9}{16}$	..	..	..	140	123	107	95	86	78	72	66	61	57	54
$\frac{5}{8}$	..	..	..	..	137	119	106	95	87	80	73	68	64	60
$\frac{11}{16}$	..	..	..	..	..	131	116	105	95	87	81	75	70	66
$\frac{3}{4}$	..	..	..	..	..	..	128	114	104	96	88	82	76	71

(142.) "*Strength of Boiler-tubes to resist External Pressure.*"—From the experiments of Fairbairn, it appears that the strength of cylindrical tubes of boiler-plate to resist collapsing pressure

varies directly as the 2·19 power of the thickness, and inversely as the diameter and length, and he gives the following rules:—

$$P = 33 \cdot 6 \times (100\,t)^{2 \cdot 19} \div (L \times d)$$
$$\text{and } p = 5 \cdot 6 \times (100\,t)^{2 \cdot 19} \div (L \times d),$$

in which P = the collapsing pressure in pounds per square inch,
p = the safe working pressure in do. do.
d = diameter of tube in inches,
L = length of tube in feet,
t = thickness of plate in inches.

To find the 2·19 power of the thickness we must use logarithms: thus, say we have a tube $\frac{1}{4}$ inch thick, 30 inches diameter, and 10 feet long. Then $100\,t$ is $\cdot 25 \times 100 = 25$, the log. of which is 1·398, and $1 \cdot 398 \times 2 \cdot 19 = 3 \cdot 0616$, the natural number due to which is 1152, and this is the 2·19 power of 25 as required; we now find P by the rule $33 \cdot 6 \times 1152 \div (30 \times 10) = 129$ lbs. per square inch.

It appears from this that $P \times d \times L$ is constant for the same thickness of plate, hence we have the rules $33 \cdot 6 \times (100\,t)^{2 \cdot 19} = P \times d \times L$; and $5 \cdot 6 \times (100\,t)^{2 \cdot 19} = p \times d \times L$.

Table 58 has been calculated by these rules, and from it we can easily find the strength of a tube by the simple rules of

TABLE 58.—For the STRENGTH of BOILER-TUBES to resist External or Collapsing Pressure.

Thickness of Plate.	Value of $P \times d \times L$ Collapsing Strain.	Value of $p \times d \times L$ Working Strain.	Thickness of Plate.	Value of $P \times d \times L$ Collapsing Strain.	Value of $p \times d \times L$ Working Strain.
in.			in.		
$\frac{1}{16}$	1860	310	$\frac{9}{16}$	228700	38120
$\frac{1}{8}$	8484	1414	$\frac{5}{8}$	288000	48000
$\frac{3}{16}$	20620	3440	$\frac{11}{16}$	354900	59150
$\frac{1}{4}$	38720	6450	$\frac{3}{4}$	429500	71600
$\frac{5}{16}$	63130	10520	$\frac{13}{16}$	511700	85300
$\frac{3}{8}$	94110	15680	$\frac{7}{8}$	601800	100000
$\frac{7}{16}$	131900	22000	$\frac{15}{16}$	700100	116700
$\frac{1}{2}$	176700	29450	1	806300	134400

arithmetic; thus for $\frac{1}{4}$-inch plate the table gives 38720 as the value of $P \times d \times L$, hence the tube 10 feet long and 30 inches diameter as above will collapse with $38720 \div (10 \times 30) =$ 129 lbs. per square inch. Again: to find the proper thickness for a flue 33 inches diameter, 20 feet long, to bear a *working* pressure of 45 lbs. per square inch; $p \times d \times L$ is in our case $45 \times 33 \times 20 = 29700$, the nearest number to which in Table 58 is 29450 opposite $\frac{1}{2}$ inch the thickness required.

(143.) From experiments on the large scale it would appear that *lap-jointed* tubes, such as are commonly used in practice, resist a collapsing strain more powerfully than the small experimental tubes from which the constants in the formula are derived. Fairbairn gives two experiments on large boilers, one 35 feet long with 3 feet 6 inch tubes, $\frac{3}{8}$ thick, which gave way with 97 lbs. per square inch, but by rule and table it should have borne $\frac{94110}{42 \times 35} = 64$ lbs. only. The other experiment was on a tube 25 feet long, 3 feet 6 inches diameter, and $\frac{3}{8}$ thick, which collapsed with 127 lbs. per square inch, but whose strength by table, &c., was $\frac{94110}{42 \times 25} = 89 \cdot 6$ lbs. only. This is only what might be expected, the double thickness at the joint acts partially like a series of rings, and increases the strength in the same way; the rules therefore give a minimum and safe strength, in fact only about 70 per cent. of the strain which ordinary boiler-tubes seem to be capable of bearing.

(144.) An obvious and easy mode of increasing the strength of a boiler-tube is by adding strong rings at intervals, so as (in effect) to divide it into short lengths, and Mr. Fairbairn proposes to do this by making the tube butt-jointed and using ⊥ iron for the junctions; but a butt-joint cannot easily be made tight, and it is better to make the tube lap-jointed as usual, and add T or L rings where required. Even this is objectionable from the great thickness of metal where the rings are riveted on, the metal there being in consequence unduly heated. An ingenious plan of Mr. Bramwell, C.E., of London, promises well to get over this objection:—Let Fig. 30 be a cylindrical tube A B C D, compressed into the ellipsis E F G H. This change of form

may be resisted in two ways, namely, by preventing E and F from giving in, or G H from giving out; if G H be prevented from bulging out, E and F will be effectually prevented from collapsing. This is effected by slipping on the tube, thin deep rings of iron, say $\frac{5}{8} \times 3$ inches deep, as J in Fig. 31; such rings are free (by their small thickness) from the objection of increasing the thickness of the metal exposed to the fire, &c., &c., to which ⊥ iron rings are liable.

Oval tubes are excessively weak in resisting an external pressure: for decidedly oval flues we have the rule :—

$$P = \frac{(100\,t)^{2 \cdot 19} \times \mathsf{M}}{(A - a) \times A \times L},$$

in which A = the major, and a = the minor axis in inches; L = the length of the tube in feet; t = the thickness in inches; P = collapsing pressure in lbs. per square inch; and M = a multiplier, which for tubes with ordinary longitudinal and cross joints = 90, and for those without cross joints = 61·4.

Mr. Fairbairn made an experiment with an oval tube $20\frac{3}{4} \times 15\frac{1}{2}$ inches, 5·083 feet long, $\frac{1}{4}$ inch thick, which collapsed with $127\frac{1}{2}$ lbs. per square inch. The rule gives :—

$$P = \frac{(100 \times \frac{1}{4})^{2 \cdot 19} \times 61 \cdot 4}{(20\frac{3}{4} - 15\frac{1}{2}) \times 20\frac{3}{4} \times 5 \cdot 083} = 127 \cdot 7 \text{ lbs.}$$

(145.) "*Factor of Safety.*"—We have admitted in (141) with Mr. Fairbairn that for ordinary cases the bursting pressure should be six times the working strain: but with that factor, 6, the thicknesses for very high pressures come out excessive and almost impracticable. Engineers have therefore been compelled to adopt a lower factor for such cases, and they do so apparently with safety. Thus, the practice of the London and North-Western Railway Company, at their Crewe works, is to use best Yorkshire plates $\frac{13}{32}$ inch thick for 4-foot boilers, with single-riveted joints, $\frac{3}{4}$ rivets, $1\frac{3}{4}$ pitch, 1 inch between rivets. Yorkshire plates, by Fairbairn's experiments, break with 42,847 lbs. per square inch of *metal between rivet-holes;* hence we have $42847 \times \frac{13}{32} \times 1 \div 1\frac{3}{4} = 9946$ lbs. per inch run of joint,

or 19,892 lbs. on the two sides. With 48 inches diameter we have $19892 \div 48 = 414$ lbs. per square inch bursting pressure of steam, and the ordinary working pressure being 120 lbs., the factor is $414 \div 120 = 3 \cdot 45$; occasionally the pressure is 150 lbs., or even more, and the factor becomes $414 \div 150 = 2 \cdot 76$.

If we admit from this that for very high pressures the factor may be as low as 4, the effect will be to add 50 per cent. to the working pressures in Table 57, &c.

CHAPTER IV.

ON THE EFFLUX OF COMPRESSED AIR, GAS, AND STEAM.

(146.) "*Efflux of Compressed Air, &c.*"—When water or other liquid escapes from an orifice in the side of a vessel into the air as at A, Fig. 32, the velocity of efflux is the same as that of a body falling freely by gravity from the height S A. Similarly when a liquid escapes from one vessel into another, by a submerged orifice B, the velocity of efflux is that due to the height S T, or the *difference* of level of the liquids in the two vessels, and it is not affected by the depth T B, at which the orifice is placed.

(147.) We have here supposed that one and the same liquid was being dealt with, but if one compartment were filled with a liquid of different specific gravity to that in the other, we have a different case. Say in Fig. 33 we have a vessel with two compartments, C and D, filled to the *same level* with two fluids, whose specific gravity (for the sake of illustration) was as 3 to 1, C being the denser of the two. The conditions of pressure at E are precisely the same as would arise with one and the same fluid in both compartments, by columns having the respective heights F E and H E, that is to say, the velocity of efflux will be that of a falling body, with the height G H, and of course it is the denser fluid which escapes with that velocity.

(148.) "*Velocity into a Vacuum.*"—Applying this reasoning to elastic fluids, we are met by the difficulty that we have no real

surface level to calculate from; in the case of air, for instance, the density diminishes as we leave the earth in geometric ratio, and the limit is in infinity. But for the purpose of calculation, we may find what the height of the atmosphere would be, if it had throughout, the same density as it has at the surface of the earth. Assuming that the barometer was at 30 inches, we find from Table 37 the density of mercury to be 13·596, water being 1; and from Table 39 air has a density of ·001221, water being 1, the height of a homogeneous column of air equal to 30 inches of mercury is therefore $\frac{13 \cdot 596 \times 30}{\cdot 001221 \times 12} = 27838$ feet, and although this is fictitious, we may use it for the purposes of calculation without error. We shall now find the velocity into a vacuum by the rule for falling bodies to be $\sqrt{27838} \times 8 = 1344$ feet per second.

(149.) The velocity of steam at atmospheric pressure into a vacuum, may be calculated in the same way: taking its density from Table 39 at ·0007613, we have $\frac{13 \cdot 596 \times 30}{\cdot 0007613 \times 12} = 44648$ feet for the height of a column of steam equivalent to 30 inches of mercury, and the velocity into a vacuum $\sqrt{44648} \times 8 = 1690$ feet per second.

Applying these rules to air, steam, &c., of other than ordinary densities and pressures, we are conducted to the remarkable fact that the velocity into a vacuum is constant, whatever the pressure may be; for instance, air of double the atmospheric pressure would have a double height of column, and thereby an increased velocity, if the density remained the same, but the density being of necessity double also, the height of column remains the same, and hence the velocity which is due to that height remains the same also. It follows from this, that if we filled a vessel with air compressed to any number of atmospheres and allowed it to escape into a vacuum, the velocity would be the same from first to last, although the pressure would be continuously reduced by the escape of the compressed air; but the *quantity* or weight of air which escapes would not be the same at all pressures, but would vary with the density of the air, which varies every moment with the pressure.

(150.) "*Velocity into Air.*"—This uniformity of velocity at all pressures does not hold when the discharge is made into air. Let Fig. 33 represent the discharge into rarefied air, say of one-third the ordinary density (made so by heat or otherwise), we have then a case analogous to (147), in which we found the velocity to be that due to the difference of height of the two columns. Thus in Fig. 33 we have two columns of the same height, or 27,838 feet, but the pressure exerted at the orifice E by the column G E is the same as that of a column of ordinary air of one-third the height, or $27838 \div 3 = 9279$ feet, and, as in (147), the velocity of efflux will be that due to the difference, or $27838 - 9279 = 18559$ feet head, namely, $\sqrt{18559} \times 8 = 1090$ feet per second. Calculating in this way, we find for pressures above atmosphere of 1, 2, 3, 4, 5, 10, 25, and 100 atmospheres, the velocity comes out 944, 1090, 1155, 1193, 1218, 1272, 1308, and 1328 feet per second respectively, which, although not uniform, is more nearly so than might have been expected. Thus, for so great a range as from 1 to 100 atmospheres, the velocity increases only $1328 \div 944 = 1{\cdot}41$, or 41 per cent.

Again, the velocity of steam of 20 lbs. pressure per square inch into the atmosphere may be calculated in the same way. By Table 38 we find that 20 lbs. per square inch is equal to 46·22 feet of water-pressure, and by Table 71 the volume of 20-lb. steam is 732 for water 1, hence the height of the column of steam generating the velocity is $46{\cdot}22 \times 732 = 33833$ feet, and the velocity of efflux $\sqrt{33833} \times 8 = 1472$ feet per second.

It will be seen from this, that for finding the velocity of efflux in any case, we require only the difference of pressure at the two sides of the orifice and the density of the issuing gas or steam; this is further illustrated by (130), &c.

(151.) When the pressure varies *very slightly*, as is usually the case in most questions of ventilation, discharge of coal-gas, &c., we may admit without sensible error that the density is constant, and the velocity of discharge will then be governed by the square root of the pressure simply, and for common air we have the rule:—

$$V = \sqrt{P} \times 66{\cdot}1$$

and for coal-gas of density ·42, air being = 1, the rule:—

$$V = \sqrt{P} \times 102,$$

in which P = pressure in inches of water, and V = velocity of discharge in feet per second: cols. 3 and 6 in Table 59 have been calculated by these rules.

TABLE 59.—Of the VELOCITY of DISCHARGE of COMMON AIR and of COAL-GAS, at ordinary Temperature and Pressure, with small differences of Pressure.

Head or Difference of Pressure in		Common Air at 62° under 30 inches of the Barometer.			Coal-gas, specific gravity ·42, that of Air being 1·0.		
		Coefficient of Discharge.			Coefficient of Discharge.		
Inches of Water.	Pounds per Square Foot.	1·0	·93	·65	1·0	·93	·65
		Velocity in Feet per Second.			Velocity in Feet per Second.		
·005	·026	4·67	4·34	3·03	7·2	6·70	4·68
·01	·052	6·61	6·14	4·29	10·2	9·48	6·63
·02	·104	9·35	8·69	6·07	14·4	13·4	9·36
·03	·156	11·4	10·6	7·41	17·6	16 3	11·4
·04	·208	13·2	12·3	8·58	20·4	19·0	13·3
·05	·260	14·8	13·7	9·62	22·8	21·2	14·8
·07	·363	17·4	16·2	11·3	27·0	25·1	17·5
·10	·519	20·9	19·4	13·6	32·3	30·0	21·0
·15	·779	25·6	23·8	16·6	39·5	36·7	25·7
·2	1·038	29·5	27·4	19·2	45·6	42·4	29·6
·25	1·298	33·1	30·8	21·5	51·0	47·4	33·1
·3	1·558	36·2	33·6	23·5	55·8	51·9	36·2
·35	1·818	39·1	36·3	25·4	60·3	56·1	39·2
·4	2·077	41·8	38·8	27·2	64·5	60·0	41·9
·45	2·337	44·3	41·2	28·8	68·4	63·6	44·5
·5	2·597	46·7	43·4	30·3	72·1	67·0	46·8
·6	3·116	51·2	47·6	33·3	79·0	73·4	51·3
·7	3·635	55·3	51·4	35·9	85·4	79·4	55·5
·8	4·155	59·1	54·9	38·4	91·2	84·8	59·3
·9	4·674	62·7	58·3	40·7	96·8	90·0	62·9
1·0	5·193	66·1	61·4	42·9	102	95	66·3
1·5	7·790	80·9	75·2	52·5	125	116	81
2·0	10·38	93·5	86·9	60·7	144	134	94
2·5	12·98	104	96·7	67·6	161	149	104
3·0	15·58	114	106	74·1	176	163	114
3·5	18·18	124	115	80·6	191	177	124
4·0	20·77	132	123	85·8	204	190	133
4·5	23·37	140	130	91·0	216	201	140
5·0	25·97	148	138	96·2	228	212	148
6·0	31·16	162	151	105·3	250	232	162

(152.) These rules may be modified so as to give the *quantity* discharged instead of the velocity; for common air the rule becomes

$$C = d^2 \times \sqrt{P} \times 21 \cdot 64,$$

and for coal-gas of density ·42 for air = 1,

$$C = d^2 \times \sqrt{P} \times 33 \cdot 4,$$

in which C = cubic feet discharged per minute, d = diameter of orifice in inches, and P = pressure in inches of water.

(153.) "*Coefficient of Contraction.*"—But the quantities discharged will vary considerably with the form of the orifice, for the issuing vein of air suffers contraction as in the case of water, and as shown by Fig. 34, where with a thin plate an orifice 1 inch diameter has the jet reduced to ·8 inch diameter, and the area to $\cdot 8^2 = \cdot 64$, that of the orifice being 1·0.

The rules in (151) give the maximum velocity, or that at the most contracted part, namely, at C; obviously the velocity at B, where the measurements are usually taken, will be less in the ratios of the respective areas; if it be 1·0 at C, in Fig. 34 it will be ·64 at B.

The experiments of Daubuisson give ·65 for an orifice in a thin plate, and ·93 for a very short cylindrical pipe, say two diameters long: cols. 4, 5 and 7, 8 in Table 59 have been calculated with those coefficients.

The more recent experiments of Poncelet, Wantzel, and St. Venant, &c., seem to show that with air and gases the coefficient is not constant for all pressures, but that it decreases as the pressure increases, so that for excess of pressure above atmosphere equal to $\frac{1}{100}$, $\frac{1}{10}$, $\frac{1}{2}$, 1, 5, 10, and 100 atmospheres, the coefficient with a thin plate becomes ·65, ·64, ·57, ·54, ·45, ·436, and ·423 respectively, and with a short tube ·834, ·82, ·71, ·67, ·54, ·51, and ·487 respectively. Table 60 has been calculated with these coefficients. A remarkable result, as applied to a tube, is that the velocity is a maximum with 5 atmospheres, becoming less with an increase of pressure beyond that point; this requires confirmation.

(154.) For steam, Péclet is of opinion that the coefficient is

constant, and may be taken at ·54 for a thin plate, and ·7 for a short tube; but the experiments in (131) show that for a thin plate Daubuisson's coefficient ·65 is correct as applied to high-pressure steam; for a short tube we may admit ·7 as correct for all pressures.

TABLE 60.—Of the VELOCITY of DISCHARGE of COMPRESSED AIR into the Atmosphere.

Excess of Pressure above the Atmosphere.			Theoretical Velocity in Feet per Second.	Thin Plate.		Short Pipe.	
In Atmospheres.	Inches of Mercury.	Lbs. per Square Inch.		Coefficient of Contraction.	Velocity in Feet per Second.	Coefficient of Contraction.	Velocity in Feet per Second.
0·01	0·3	0·147	132	·65	85·8	·834	110
0·10	3·0	1·47	402	·64	257	·82	330
0·5	15	7·35	771	·57	439	·71	547
1·0	30	14·7	944	·54	510	·67	632
5·0	150	73·5	218	·45	548	·54	(658)
10·0	300	147·0	1272	·436	555	·51	649
100·0	3000	1470·0	1328	·423	562	·487	646
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)

(155.) "*Friction of Long Pipes.*"—We have so far considered only the head or pressure necessary to give any required velocity with an orifice, or a pipe so short that the friction was inappreciable; but where the length is considerable, there is a second loss of pressure due simply to the *friction* of the air or steam against the sides of the pipe; and in all such cases the head due both to velocity and to friction must be separately calculated and the sum total taken.

(156.) The head due to friction alone may be calculated by the following rules:—

$$H = C^2 \times L \div (3 \cdot 7\, d)^5$$

$$C = \left(H \times (3 \cdot 7\, d)^5 \div L\right)^{\frac{1}{2}}$$

$$d = \left(C^2 \times L \div H\right)^{\frac{1}{5}} \div 3 \cdot 7$$

$$L = (3 \cdot 7\, d)^5 \times H \div C^2,$$

in which d = the diameter of the pipe in inches, L = the length of pipe in yards, C = cubic feet per minute, and H = the head

TABLE 61.—Of the FRICTION of AIR, STEAM, and GAS in LONG PIPES.

Diameter of Pipe in Inches.	Head, or Difference of Pressure at the two ends of a Pipe 1 Yard long, in Inches of Water.	Head, or Difference of Pressure at the two ends of a Pipe 1 Yard long, in Pounds per Square Inch.
	Head for 1 Cubic Foot per Minute.	
1/4	1·477	·05317
3/8	·1945	·00700
1/2	·0185	·000666
3/4	·006077	·0002187
1	·001442	·0000519
	Head for 10 Cubic Feet per Minute.	
1 1/4	·04725	·001701
1 1/2	·01899	·000684
1 3/4	·00878	·000326
2	·00451	·000162
2 1/4	·00250	·000090
2 1/2	·00148	·0000532
	Head for 100 Cubic Feet per Minute.	
3	·0594	·00214
3 1/2	·0746	·000989
4	·0141	·000507
4 1/2	·00782	·000281
5	·00462	·000166
6	·00186	·0000668
7	·000858	·0000309
8	·000440	·0000158
	Head for 1000 Cubic Feet per Minute.	
9	·02442	·000879
10	·01442	·000519
12	·00579	·000209
15	·00189	·0000681
18	·000763	·0000275
21	·000353	·0000127
24	·000181	·00000653
	Head for 10,000 Cubic Feet per Minute.	
30	·00593	·000214
33	·003685	·000133
36	·002385	·0000858
42	·001103	·0000398
48	·000566	·0000204
54	·000314	·0000113
60	·000185	·00000667

(or difference of pressure at the two ends of the pipe) in inches of water. These rules cannot be easily worked without the use of logarithms; but we have calculated by them Table 61, for the use of which we have the following rules:—

(157.) 1st. Having the diameter, length, and discharge given, to find the head, take from Table 61, opposite the given diameter, the number in column 2 or 3 (according to the terms in which the head is desired), and multiply it by the square of the given discharge in cubic feet, and by the length in yards; and the product is the head in inches of water, or in pounds per square inch, due to friction alone, to which the head due to velocity has to be added from Tables 59 or 119, &c.

(158.) 2nd. Having the head, diameter, and length given, to find the discharge, assume a discharge and calculate the head for that as in (157); divide the assumed discharge by the square root of the head due to it, and multiply by the square root of the given head; and the quotient is the true discharge sought.

(159.) In order to facilitate calculation, the Table 61 is so arranged, that for diameters under one inch the discharge must be taken in cubic feet, as in (157), &c., but from $1\frac{1}{4}$ inch to $2\frac{1}{2}$ inches in tens, from 3 inches to 8 inches in hundreds, from 9 inches to 24 inches in thousands, and from 30 to 60 inches in tens of thousands of cubic feet.

Thus a $\frac{3}{4}$-inch pipe, 20 yards long, discharging 2 cubic feet of air per minute, requires $\cdot 006 \times 2^2 \times 20 = \cdot 48$ inch head of water for friction, and the velocity being $\dfrac{144 \times 2}{\cdot 44 \times 60} = 10 \cdot 9$ feet per second, the head for that velocity by Table 59 is $\cdot 03$ inch of water, making $\cdot 48 + \cdot 03 = \cdot 51$ inch total.

A $2\frac{1}{2}$-inch pipe, 40 yards long, with 38 cubic feet of air, will require $\cdot 00148 \times \left(\dfrac{38}{10}\right)^2 \times 40 = \cdot 85$ inch head for friction, and the velocity being $\dfrac{144 \times 38}{4 \cdot 9 \times 60} = 18 \cdot 6$ feet per second, or by Table 59 $= \cdot 1$ inch head, we have a total of $\cdot 85 + \cdot 1 = \cdot 95$ inch head.

A 4-inch pipe, 10,560 yards long, discharging 852 cubic feet

of *gas* per hour, or 14·2 cubic feet per minute, will require $\cdot 0141 \times \left(\frac{14\cdot 2}{100}\right)^2 \times 10{,}560 = 3\cdot 002$ inches head for friction. With so small a discharge, the head for velocity will be inappreciable; Clegg found by experiment, that the head with a pipe having these conditions was 3 inches of water.

A 9-inch pipe, 30 yards long, discharging 500 cubic feet of gas per minute, will require $\cdot 02442 \times \left(\frac{500}{1000}\right)^2 \times 30 = \cdot 183$ inch head for friction, and the velocity being $\frac{144 \times 500}{63\cdot 6 \times 60} = 18\cdot 8$ feet per second, the head for that velocity by Table 59 is ·04 inch, or $\cdot 183 + \cdot 04 = \cdot 223$ inch total.

(160.) Again, say we require to know the discharge of 45 lbs. steam by a 4-inch pipe, 150 yards long, with a loss of 1 lb. in pressure, the steam at the exit end being reduced to 44 lbs. per square inch. Let us assume, say 400 cubic feet for the discharge, and calculating as in (158), we have $\cdot 000507 \times \left(\frac{400}{100}\right)^2 \times 150 = 1\cdot 22$ lb. per square inch, as the head due to friction alone. The area of a 4-inch pipe being 12·56 square inches, or $12\cdot 56 \div 144 = \cdot 087$ square foot, the velocity must be $400 \div \cdot 087 = 4600$ feet per minute, or 76 feet per second, which at the contraction (154) with ·7 for coefficient becomes $76 \div \cdot 7 = 109$ feet per second, the head due to which is $(109 \div 8)^2 = 185$ feet of 45 lbs. steam, or $185 \div 439 = \cdot 42$ foot water, or $\cdot 42 \div 2\cdot 3 = \cdot 18$ lb. per square inch for velocity alone. Thus for our *assumed* quantity the total head is $1\cdot 22 + \cdot 18 = 1\cdot 4$ lb. per square inch instead of 1 lb. the pressure allowed in our case. The true discharge with the given head is therefore $400 \times \sqrt{1} \div \sqrt{1\cdot 4} = 339$ cubic feet of steam per minute (158).

(161.) This last result is obtained by the application of a useful general law, which may be stated thus: the discharge of any pipe or system of pipes, apertures, &c., &c., is proportional to the square root of the head; and conversely, the head is proportional to the square of the discharge.

(162.) "*Square and Rectangular Channels.*"—The case of

square and other rectangular channels may be assimilated to that of round pipes, and then the velocity, &c., may be calculated by the rules and tables given for the latter. The velocity of discharge, whatever may be the form of the pipe or channel, is in all cases proportional to the sectional area divided by the periphery or circumference. In round pipes, this is always one-fourth of the diameter: thus a pipe 1 inch diameter has an area of $\cdot 7854$, and a circumference of $3 \cdot 142$, and $\cdot 7854 \div 3 \cdot 142 = \cdot 25$; and a 4-inch pipe gives $12 \cdot 57 \div 12 \cdot 57 = 1$.

"*Square Pipes.*"—Square pipes give the same uniform ratio: thus a pipe 1 inch square will have an area of 1, and a periphery of 4 inches, and $\frac{1}{4} = \cdot 25$ as with a circular pipe: again, a pipe 4 inches square has an area of 16 square inches, and a periphery of 16 inches also, and as with the round pipe $\frac{16}{16} = 1$. It follows that the *velocity* of discharge with a square pipe is the same as with a round one, with the same length and head, &c.; but of course the quantity discharged will be greater with a square pipe in the same proportion, as the area of a square is greater than that of a circle, or as 1 to $\cdot 7854$.

"*Rectangular Pipes.*"—The same laws apply to rectangular pipes: thus a pipe 6 inches by 3 inches has an area of 18, and a periphery of 18 also, and $\frac{18}{18} = 1$, which is the same as a 4-inch pipe, as we have seen; therefore a round pipe 4 inches diameter will have the same velocity of discharge as a pipe 4 inches square, or as another 6 inches $\times$ 3 inches, and the quantities discharged will be in proportion to the areas, or $12 \cdot 57$, 16, and 18 respectively.

Say we have a rectangular channel, 40 yards long, 36 inches wide, and 18 inches deep, and we require to know the head or pressure for a velocity of 6 feet per second. The area is $36 \times 18 = 648$, and the periphery 108, and we have $648 \div 108 = 6$, which is the same as a pipe $6 \times 4 = 24$ inches diameter, and we can calculate the head as for a round pipe of that diameter. A round 24-inch or 2-feet pipe, discharging at the rate of 6 feet per second, would deliver $3 \cdot 14 \times 6 \times 60 = 1130$ cubic feet per minute, requiring $\cdot 000181 \times \left(\frac{1130}{1000}\right)^2 \times 40 = \cdot 0077$ inch head for friction alone. The head for 6 feet velocity by Table 59

with $\cdot 93$ for coefficient, is $\cdot 01$ inch of water; the total head is therefore $\cdot 0077 + \cdot 01 = \cdot 0177$ inch of water, and this is also the head for a pipe 36 inches by 18 inches, as in our case.

(163.) "*Effect of Repeated Enlargements and Contractions.*"—It might be supposed, that the effect of enlarging the channel would be to diminish friction in discharging a fixed quantity of air, and this is true where the velocity has not to be got up again; but where there are repeated and successive contractions and enlargements, the head saved in friction by each enlargement, may be more than compensated by that required for getting up the velocity at the next contraction. Let Fig. 113 represent the rectangular pipe 40 yards long, the head for which with a velocity of 6 feet per second we have just calculated to be $\cdot 0177$ inch of water.

Let Fig. 114 be a similar pipe, but one having two chambers or rooms, A and B, in its course, by which the length of the *pipe* or channel itself, C F, is reduced to one-half or 20 feet. The velocity of the air passing through the two rooms is so very small, that there will practically be no friction there, so that the friction is thus reduced to one-half also. But we found that the head at C, due to the air entering the pipe with 6 feet velocity, was $\cdot 01$; when the air enters the room A, that velocity is lost, and must be got up again for the air entering the next contraction at D, to be again lost in B, and got up again at E. The head for friction in this case is $\cdot 0077 \div 2 = \cdot 00385$, and the head for the velocity at 3 places $\cdot 01 \times 3 = \cdot 03$, making a total of $\cdot 00385 + \cdot 03 = \cdot 03385$, or about double the head for a uniform pipe; so far therefore from diminishing the head by enlarging the channel at A and B, we have really doubled it in this case. It is important to keep this fact in view in cases of ventilation, &c.; in large buildings, the changes of area in the passages are numerous and unavoidable, they are also too complicated to be calculated, but in the case of the Prison Mazas (394), we found that the velocity was thus reduced to $\cdot 423$ of the theoretical velocity, and in the Prison of Provins (397), to $\cdot 322$.

(164.) "*Discharge of Steam.*"—It is sometimes convenient to

estimate the discharge of steam by the horse-power instead of by the volume, although, as we have shown (117), the latter is by far the most satisfactory. Admitting that 1 cubic foot of water evaporated to steam at any pressure (19) is equal to 1 nominal horse-power (118), Table 71 gives direct the volume of steam per cubic foot of water for the different pressures: thus, with 45 lbs. steam, we have 439 cubic feet of steam per cubic foot of water, or per horse-power. Say that we have 100 horse-power and a short 4-inch pipe, and require the loss of pressure necessary with *uniform velocity* (166). We have $439 \times 100 \div 3600 = 12 \cdot 2$ cubic feet per second; the area of the 4-inch pipe is $12 \cdot 56$ square inches, which at the contraction becomes $12 \cdot 56 \times \cdot 7 \div 144 = \cdot 061$ square foot, hence the velocity is $12 \cdot 2 \div \cdot 061 = 200$ feet per second, the head due to which by the laws of falling bodies is $(200 \div 8)^2 = 625$ feet, which being a column of 45 lbs. steam, is equal to $625 \div 439 = 1 \cdot 424$ foot of water, or $1 \cdot 424 \div 2 \cdot 3 = \cdot 619$ lb. per square inch; the pressure at entry being 45 lbs., that at the exit end of the short pipe will be $45 - \cdot 619 = 44 \cdot 381$ lbs.

(165.) The pressure thus lost in discharging a fixed volume of steam varies inversely as the fourth power of the diameter of the orifice, for this reason: the area of a circular orifice, and consequently the velocity of efflux with a fixed quantity varies as d^2, and as the pressure varies as V^2, the ratio becomes $d^2 \times V^2$, or more simply as d^4. For instance, if in any particular case we reduced the diameter to one-half, the area would be reduced to one-fourth, therefore the velocity necessary for a fixed quantity must be increased in the ratio 1 to 4, and the pressure to generate that velocity, in the ratio 1 to 4^2 or 1 to 16, so that with diameters in the ratio 1 to 2, the pressures are in the ratio 1 to 2^4 or 1 to 16. The pressure with a 4-inch pipe being $\cdot 619$ lb., that with say a 6-inch pipe will be $\cdot 619 \times 4^4 \div 6^4$, or $\cdot 619 \times 256 \div 1296 = \cdot 122$ lb., &c.

(166.) "*Steam-pipes to Engines.*"—In applying these rules to pipes for steam-engines, it must be observed that the supply of steam to an ordinary engine is intermittent, being 0 when passing the centre, and a maximum when at the middle of the stroke, and that the maximum velocity should be taken in calcu-

lating the size of steam-pipe. Thus, an engine of 5 feet stroke and 22 revolutions per minute has a *mean* velocity of $5 \times 2 \times 22 = 220$ feet per minute; but when at the centre of the stroke, the piston is moving with the velocity of the crank-pin, or $5 \times 3 \cdot 14 \times 22 = 345$ feet per minute, instead of 220. It will be found that in a common double-acting engine the maximum velocity is $1 \cdot 57$ times the mean velocity, and with a pair of engines having their cranks at right angles the ratio is $1 \cdot 11$ to 1: so that, for instance, one engine of 100 horse-power takes steam at the maximum rate of 157 horse-power, and a pair of engines, each of 50 horse-power, combined at right angles, takes 111 horse-power of steam at the maximum speed. Applying this to the 4-inch pipe in (164), 100 horse-power with uniform velocity would be equivalent to the maximum velocity with a single steam-engine of $100 \div 1 \cdot 57 = 64$ horse-power, or that of a pair of engines of the collective power of $100 \div 1 \cdot 11 = 90$ horses. Tables 119 and 120 have been calculated in this way.

(167.) Table 120 gives the friction in steam-pipes calculated by these rules, with the correction explained in (166) as applied to steam-engines where the velocity is variable. This table in connection with Table 119 will enable us to calculate easily the sizes of steam-pipes in most ordinary cases: the following examples will illustrate their application.

Example 1.—A single engine of 10 horse-power, with a 1½-inch steam-pipe 6 yards long, and 45 lbs. steam, will by Table 119 require $\cdot 704$ lb. for velocity at entry, and by Table 120, $\cdot 082 \times 6 = \cdot 492$ lb. for friction, making a total of $\cdot 704 + \cdot 492 = 1 \cdot 196$ lb. per square inch, so that the pressure available for working the engine is reduced to $45 - 1 \cdot 196 = 43 \cdot 8$ lbs. per square inch.

Example 2.—A single engine of 100 horse-power, with a 5-inch pipe 12 yards long, and 25 lbs. steam, will require for velocity $1 \cdot 0$ lb., and for friction $\cdot 0554 \times 12 = \cdot 66$ lb., making a total of $1 \cdot 0 + \cdot 66 = 1 \cdot 66$ lb., and the pressure at the engine is reduced to $25 - 1 \cdot 66 = 23 \cdot 34$ lbs. per square inch.

Example 3.—A pair of low-pressure engines of the collective power of 1000 horses, with a 16-inch steam-pipe 10 yards long, and 7 lbs. steam in the boiler, will by Table 119 require $\cdot 783$ lb.

for velocity, and by Table 120 (say for 1041 horse-power) $\cdot 0238 \times 10 = \cdot 238$ lb. for friction, making a total of $\cdot 783 + \cdot 238 = 1 \cdot 031$, say 1 lb. per square inch, thus reducing the effective pressure at the engine to $7 - 1 = 6$ lbs. per square inch.

Example 4.—Say that the steam of a 20-horse boiler is used for evaporating pans or similar work where the velocity is uniform; then with 25 lbs. steam, and 50 yards of 2-inch pipe, Table 119 gives $\cdot 618$ lb. for velocity, and Table 120, $\cdot 0779 \times 50 = 3 \cdot 9$ lbs. for friction, making a total of $\cdot 618 + 3 \cdot 9 = 4 \cdot 518$ lbs., and the pressure is reduced to $25 - 4 \cdot 5 = 20 \cdot 5$ lbs. per square inch.

(168.) When the diameter of the piston, &c., is known, it is better to calculate for the volume of steam rather than by the horse-power (117). Thus, say we have an engine with 24-inch cylinder, 4 feet stroke, making 30 revolutions per minute, the steam-pipe being 4 inches diameter, 20 yards long, and the pressure of steam in the boiler 25 lbs. per square inch.

The *maximum* velocity of the piston is that of the crank-pin, or $4 \times 3 \cdot 14 \times 30 = 377$ feet per minute $= 6 \cdot 3$ feet per second, which with a 4-inch pipe becomes $6 \cdot 3 \times 24^2 \div 4^2 = 226$ feet per second, or with $\cdot 7$ for coefficient, $226 \div \cdot 7 = 323$ feet per second in the contraction at entry (153), the head due to which is $(323 \div 8)^2 = 1632$ feet of 25 lbs. steam, and the volume of steam at that pressure being by Table 71 $= 644$ times that of water, we have $1632 \div 644 = 2 \cdot 53$ feet of water, or $2 \cdot 53 \div 2 \cdot 3 = 1 \cdot 1$ lb. per square inch as the pressure due to velocity only.

Then the area of the piston being $3 \cdot 14$ square feet, we have $3 \cdot 14 \times 377 = 1184$ cubic feet per minute as the *maximum* quantity passing through the pipe, and by Table 61 we shall have for friction alone $\cdot 000507 \times \left(\frac{1184}{100}\right)^2 \times 20 = 1 \cdot 42$ lb. per square inch, which added to the head due to velocity makes the total loss of pressure $1 \cdot 1 + 1 \cdot 42 = 2 \cdot 52$ lbs. per square inch, and the effective pressure at the engine is reduced to $25 - 2 \cdot 52 = 22 \cdot 48$ lbs. per square inch.

CHAPTER V.

CHIMNEYS.

(169.) "*Chimneys to Steam-boilers.*"—The case of a chimney is analogous to that in (147) and (150), where we had two columns of equal height but unequal density, and we found that the velocity was that due to the reduced difference of the two columns. It has been shown in (100) that the temperature of the air, &c., in well-arranged steam-boiler chimneys is about 550°, and by Table 24 it will be found that at that temperature the density of air is almost exactly one-half of the density at 62°, and we have, as in Fig. 35, a column of air in the chimney, say 80 feet high, weighing only half as much as the same height of external air, and motion will ensue as in (150) with a velocity due to 80 − 40 = 40 feet head. For the purposes of calculation this head may be represented in inches of water, thus the density of air at 62° being $\frac{1}{820}$th that of water, and at 552° $\frac{1}{1640}$th, a column 80 feet or 960 inches high will be equal to $\frac{960}{820} = 1 \cdot 17$ and $\frac{960}{1640} = \cdot 585$ inch respectively, and the difference is $1 \cdot 17 - \cdot 585 = \cdot 585$ inch of water, see (221).

Assuming a fixed or standard temperature for the chimney at 552°, we have in Table 62 the equivalent differences of

TABLE 62.—Of the DRAUGHT POWERS of CHIMNEYS, &c., with the Internal Air at 552°, and External Air at 62°, and with the Damper nearly closed.

Height of Chimney in Feet.	Draught Power in Inches of Water.	Theoretical Velocity in Feet per Second.		Height of Chimney in Feet.	Draught Power in Inches of Water.	Theoretical Velocity in Feet per Second.	
		Cold Air Entering.	Hot Air at Exit.			Cold Air Entering.	Hot Air at Exit.
10	·073	17·8	35·6	80	·585	50·6	101·2
20	·146	25·3	50·6	90	·657	53·7	107·4
30	·219	31·0	62·0	100	·730	56·5	113·0
40	·292	35·7	71·4	120	·876	62·0	124·0
50	·365	40·0	80·0	150	1·095	69·3	138·6
60	·438	43·8	87·6	175	1·277	74·8	149·6
70	·511	47·3	94·6	200	1·460	80·0	160·0

pressure in inches of water, by which we may calculate the velocities, &c., by the rules and Tables 59, 61, &c., &c.

We allowed in (98) 300 cubic feet of air at 62° per pound of coal; in passing through the fire this is highly heated and it leaves at 1200°, and is expanded to about $3\frac{1}{2}$ times its former volume (Table 24); from thence to the chimney it is progressively cooled to 552°, and becomes reduced to double its normal volume, or to 600 cubic feet. If we allow 10 lbs. of coal per horse-power, we have to pass 6000 cubic feet up the chimney per horse-power per hour.

(170.) "*Round Chimneys.*"—Say we require the power of a chimney 80 feet high, 2 feet 9 inches diameter, attached to steam-boilers 30 feet long, having flues the same area as the chimney, and say 100 feet long in circuit from furnace to base of chimney. It will be seen that we have to determine the discharge of a pipe 180 feet or 60 yards long, 2 feet 9 inches diameter, with a head of ·585 inch of water by Table 62 and (169). We must assume a discharge as in (158), say 100 horse-power or $100 \times 6000 \div 60 = 10000$ cubic feet per minute, which by Table 61 will require $\cdot003685 \times 60 = \cdot2211$ inch of water for friction alone; we have to add to this the head due to velocity. The diameter being 2·75 feet, we have an area of 5·94 feet, and as we have $10000 \div 60 = 167$ cubic feet per second, the velocity will be $167 \div 5\cdot94 = 28$ feet per second, which by Table 59, with ·93 coefficient, is due to a head of ·2 inch of water, and the total head for 100 horse-power is $\cdot2211 + \cdot2 = \cdot4211$ inch of water. We have, however, ·585 inch at disposal, and by (158) or (161) this will be equal to $\dfrac{100 \times \sqrt{\cdot585}}{\sqrt{\cdot4211}}$ or $\dfrac{100 \times \cdot765}{\cdot649} = 118$ horse-power. A chimney of these dimensions is working well at Dartford, the consumption of coal being 10 cwt. per hour, which allowing 10 lbs. per horse-power, as we assumed in (169), is equal to $1120 \div 10 = 112$ horse-power.

(171.) "*Square Chimneys.*"—If the chimney we have just considered had been square, the horse-power would have been greater in the simple proportion of the areas of a square to a

circle, see (162). The *velocity* of discharge is the same in both cases, and the quantity discharged is proportional to the respective areas, or as ·7854 to 1; in our example (170), a square chimney would have been equal to 118 ÷ ·7854 = 150 horse-power.

(172.) We shall assume a constant length of circuit of flues at 100 feet; this will be too great for small boilers, but the only effect will be to make the chimney rather too powerful for such cases. It is expedient to allow a margin for unforeseen contingencies, and to give an excess of power in all cases; the damper can be regulated so as to obviate any mischievous results from the admission of an excess of air (108). We have calculated Table 63 on these principles, giving the power throughout at 75 per cent. of the maximum calculated power, thus allowing 25 per cent. for margin.

Figs. 44 to 46 give elevations of common chimneys of 40, 60, and 80 feet height; care should be taken not to contract the channel at the points B, C, D, to less area than the outlet A at the top. Mortar should be used for the most part, because cement is destroyed by a strong heat; the 4½-inch work at the top, however, should be in good cement; with so thin a wall the heat is rapidly carried off by the external air, and the cement will not be injured. With steam-boilers the heat of the air should not exceed 600° (100), and ordinary stock-bricks will stand that temperature well, but with reverberatory and other brick furnaces (87), the air is at a temperature of about 2250°, and for such cases the chimney should be lined with fire-brick throughout, and as the cohesion of mortar is soon destroyed with such high temperatures, there should be wrought-iron bands round the outside at regular distances from top to bottom. In ordinary chimneys hoop-iron should be built into the brickwork every few courses to form a bond; and a lightning conductor should not be omitted.

(173.) "*Effect of Long and Short Flues.*"—The effect of different lengths of flue is shown by Table 64, in which we have taken as an example a chimney 60 feet high and 2 feet 9 inches square, which by Table 63 with an ordinary flue 100 feet long, is equal to 100 horse-power; it will be seen

TABLE 63.—Of the POWER of CHIMNEYS to STEAM-BOILERS, having Flues 100 feet long, in circuit from Furnace to base of Chimney.

Size at the Top, inside.	40 Feet.		60 Feet.		80 Feet.		100 Feet.		120 Feet.		160 Feet.	
	Round.	Square.	Round.	Square.	Round.	Square.	Round.	Square.	Round.	Square.	Round.	Square.
ft. in.	H.P.	H.P.										
1 0	6·4	8·1										
1 3	10·9	13·9	12·8	16·3								
1 6	16·6	21·0	19·5	24·8	21·7	27·5						
1 9	23·6	30·0	27·9	34·2	31·1	40·0						
2 0	31·9	41·0	37·3	47·5	42·3	53·8	45·7	58·2				
2 3	..	..	49·4	62·8	55·3	70·4	60·0	76·4	63·8	81·2		
2 6	..	..	65·3	83·1	70·4	90·0	76·5	97·4	81	103	85	108
2 9	..	..	78	100·0	88	112	94·9	121	101	128	106	135
3 0	..	..	94	123	106	135	114	145	123	157	130	165
3 6	..	..	..	..	150	191	163	207	175	223	186	237
4 0	..	..	..	..	202	257	220	280	235	300	252	321
5 0	..	..	..	..	..	..	360	458	388	494	415	528
6 0	..	..	..	..	..	..	..	..	577	734	615	783

NOTE.—The power of the chimneys in this table is three-fourths of their absolute maximum power; thus the maximum power of a chimney 3 feet 6 inches diameter, 80 feet high, is $\frac{150 \times 4}{3}$ = 200 horse-power, &c.

that with a flue of one-half the length, or 50 feet, the power is increased to 107·6 horse-power only, and that with a flue 1000 feet long, the power is reduced to one-half nearly. This may be applied to other cases; say we required a chimney of 150 horse-power, with a flue 1000 feet long (from furnace to chimney), this would be equal to 150 ÷ ·514 = 300 horse-power in Table 63, and might be 120 feet high and 4 feet square. Again, a chimney of 50 horse-power, with a flue 400 feet long, must be equal to 50 ÷ ·708 = 70 horse-power, in Table 63, and may be 80 feet high, and either 2 feet 6 inches round, or 2 feet 3 inches square, &c., &c.

TABLE 64.—Of the POWER of a CHIMNEY 60 feet high, 2 feet 9 inches square, with Flues of different Lengths.

Length of Flue in Feet.	Horse-power.	Length of Flue in Feet.	Horse-power.
50	107·6	800	56·1
100	100·0	1000	51·4
200	85·3	1500	43·3
400	70·8	2000	38·2
600	62·5	3000	31·7

(174.) "*Effect of Internal Temperature in Chimneys.*"—The discharging power of a chimney increases with increase of internal temperature, but not to an unlimited extent, for while the draught power increases, so does the volume of air due to a given weight increase by expansion, and the result is that the *weight* of air discharged attains a maximum at a certain temperature, and an increase beyond that point results in a diminution of effect.

This temperature is 525° according to Péclet, but this is true only for cases such as a reverberatory furnace, where the fire escapes direct into the chimney, and the flue being very short, friction may be neglected, and the whole power of the chimney is expended in generating velocity. Table 65 shows in col. 6 that the velocity, and therefore the *weight* of cold air, is then a maximum at 582 − 62 = 522° above the external air. But the weight of air necessary to carry off the heat (84), increases

rapidly as the temperature is reduced, as shown by col. 8, and as a result, the power of the chimney as measured by the consumption of fuel, increases with the temperature throughout, col. 9.

TABLE 65.—Of the POWER, with different INTERNAL TEMPERATURES, of a CHIMNEY 32 feet high, with a very short Flue, as in Reverberatory and other Furnaces: External Air 62°.

Volume of Air in the Chimney: External Air = 1·0.	Temp. of Air in the Chimney.	Draught in Inches of Water.			Velocity of Air in Feet per Second.		Pounds of Air per lb. of Coal.	Pounds of Coal per Sq. Foot of Chimney per Hour.
		For Velocity of Cold Air at Entry.	For *Extra* Velocity of Hot Air at Exit.	Total.	Cold Air at Entry.	Hot Air at Exit.		
1·25	° 192	·0891	·0045	·0936	19·73	24·69	400	14
1·5	322	·1328	·0222	·155	24·10	36·15	200	33
1·75	452	·1514	·0486	·200	25·72	45·00	133	53
2·00	582	·1570	·0790	·236	26·30	52·60	100	72
2·25	712	·1530	·1070	·260	25·99	58·44	80	89
3·0	1102	·1337	·1783	·312	24·18	72·50	50	133
4·0	1622	·1080	·2430	·351	21·70	86·90	33	178
5·0	2142	·0890	·2850	·374	19·75	98·75	25	217
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)

(175.) With chimneys to steam-boilers, the friction of the long flues must be considered as well as the head due to velocity, the result being that the maximum effect is attained when the volume of the internal air is between three and four times that of the external air, as shown by col. 5 of Table 66. If we admit 3½ as the relative volume, the temperature would be about 1300° by Table 24; but with such a high temperature there would be an enormous loss of useful effect, and the power of the chimney would be only 116 ÷ 107 = 1·085, or 8·5 per cent. greater than with double volume, which experience has shown (100) to be the best in practice. The table shows, however, that a variation of 130° either way has little influence on the power of the chimney; thus with volume 1¾ we have 100 ÷ 107 = ·9346, or ·0654 = 6·54 per cent. less, and with volume 2¼, 112 ÷ 107 = 1·047, or 4·7 per cent. greater power than with volume 2.

Col. 5 of Table 66 gives the *ratio* of the power of the chimney at the different internal temperatures; and col. 6, the *maximum* 172) consumption of fuel per square foot.

TABLE 66.—Of the POWER, with different INTERNAL TEMPERATURES, of a CHIMNEY 80 feet high, 2 feet 9 inches diameter, with a Flue of the same Area 100 feet long from Furnace to Foot of Chimney.

Volume of Air in the Chimney: External Air = 1·0.	Temperature of		Draught of Chimney in Inches of Water.	Ratio of the Power at different Temperatures.	Pounds of Coal per Square Foot of Chimney per Hour.
	Air in the Chimney and Flue.	External Air.			
	°	°			
1·25	192	62	·234	71	120
1·5	322	62	·390	89	150
1·75	452	62	·500	100	168
2·00	582	62	·585	107	180
2·25	712	62	·650	112	188
3·0	1102	62	·780	116	195
4·0	1622	62	·890	116	195
5·0	2142	62	·926	114	192
(1)	(2)	(3)	(4)	(5)	(6)

CHAPTER VI.

ON VAPOURS.

(176.) "*Elastic Force of Vapour.*"—Let Fig. 57 be a barometer with a large chamber A, one cubic foot in capacity, and let the bore of the tube be very small, so that its capacity may be regarded as infinitely small compared with that of the chamber; the space A will then be a perfect vacuum, and the height of the column of mercury at ordinary atmospheric pressure will be say 30 inches. Now if a few drops of water be introduced into A, vapour will instantly be formed from it, filling the chamber and depressing the column B say to G, the amount of this depression will depend on the temperature of A; by adding mercury at C until that column is raised say to D, the column B may be restored to its former level. Now the height of the column C B was the measure of the atmospheric pressure, and as that is supposed to be constant, it is evident that a pressure equal to C D is exerted by the vapour in the chamber A; thus if C D is 1 inch, D B will then be 29 inches only, and as 30 is required to balance the atmospheric pressure, the rest must have

Table 67.—Of the Elastic Force of Vapour of Water in Inches of Mercury, calculated from the Experiments of Regnault.

Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.
°		°		°		°		°	
0	·044	44	·288	88	1·323	132	4·752	176	13·96
1	·046	45	·299	89	1·366	133	4·881	177	14·28
2	·048	46	·311	90	1·401	134	5·012	178	14·60
3	050	47	·323	91	1·455	135	5·146	179	14·93
4	·052	48	·335	92	1·501	136	5·283	180	15·27
5	·054	49	·348	93	1·548	137	5·423	181	15·61
6	·057	50	·361	94	1·596	138	5·565	182	15·96
7	·060	51	·374	95	1·646	139	5·710	183	16·32
8	·062	52	·388	96	1·697	140	5·858	184	16·68
9	·065	53	·403	97	1·751	141	6·010	185	17·05
10	·068	54	·418	98	1·806	142	6·165	186	17·42
11	·071	55	·433	99	1·862	143	6·324	187	17·81
12	·074	56	·449	100	1·918	144	6·488	188	18·20
13	·078	57	·465	101	1·976	145	6·655	189	18·60
14	·082	58	·482	102	2·036	146	6·825	190	19·01
15	·086	59	·500	103	2·098	147	7·000	191	19·42
16	·090	60	·518	104	2·162	148	7·177	192	19·83
17	·094	61	·537	105	2·227	149	7·360	193	20·26
18	·098	62	·556	106	2·293	150	7·546	194	20·68
19	·103	63	·576	107	2·361	151	7·736	195	21·12
20	·108	64	·596	108	2·431	152	7·930	196	21·57
21	·113	65	·617	109	2·503	153	8·128	197	22·03
22	·118	66	·639	110	2·577	154	8·330	198	22·50
23	·123	67	·661	111	2·653	155	8·536	199	22·97
24	·129	68	·685	112	2·731	156	8·746	200	23·46
25	·135	69	·708	113	2·811	157	8·960	201	23·95
26	·141	70	·733	114	2·893	158	9·177	202	24·45
27	·147	71	·759	115	2·977	159	9·400	203	24·95
28	·153	72	·785	116	3·063	160	9·628	204	25·47
29	·160	73	·812	117	3·151	161	9·861	205	26·00
30	·167	74	·840	118	3·241	162	10·099	206	26·53
31	·174	75	·868	119	3·333	163	10·342	207	27·08
32	·181	76	·897	120	3·427	164	10·590	208	27·64
33	·188	77	·927	121	3·523	165	10·843	209	28·19
34	·196	78	·958	122	3·621	166	11·101	210	28·76
35	·204	79	·990	123	3·721	167	11·36	211	29·34
36	·212	80	1·023	124	3·824	168	11·63	212	29·92
37	·220	81	1·057	125	3·930	169	11·90	213	30·52
38	·229	82	1·092	126	4·039	170	12·18	214	31·13
39	·238	83	1·128	127	4·151	171	12·46	215	31·75
40	·247	84	1·165	128	4·265	172	12·75	216	32·38
41	·257	85	1·203	129	4·382	173	13·05	217	33·02
42	·267	86	1·242	130	4·502	174	13·35	218	33·67
43	·277	87	1·282	131	4·625	175	13·66	219	34·33

TABLE 67.—*continued.*

Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.	Temp. Fahr.	Force in Inches.
°		°		°		°		°	
220	35·01	266	79·93	312	163·2	358	304·7	404	527·6
221	35·68	267	81·27	313	165·6	359	308·6	405	533·5
222	36·37	268	82·63	314	168·0	360	312·6	406	539·5
223	37·08	269	84·01	315	170·5	361	316·5	407	545·5
224	37·80	270	85·41	316	172·9	362	320·5	408	551·6
225	38·53	271	86·83	317	175·4	363	324·6	409	557·7
226	39·27	272	88·26	318	177·9	364	328·7	410	563·9
227	40·02	273	89·72	319	180·5	365	332·8	411	570·1
228	40·79	274	91·18	320	183·1	366	336·9	412	576·4
229	41·56	275	92·67	321	185·7	367	341·2	413	582·8
230	42·34	276	94·18	322	188·3	368	345·4	414	589·2
231	43·14	277	95·72	323	191·0	369	349·7	415	595·7
232	43·95	278	97·27	324	193·7	370	354·0	416	602·2
233	44·78	279	98·84	325	196·4	371	358·4	417	608·8
234	45·62	280	100·4	326	199·2	372	362·8	418	615·4
235	46·47	281	102·0	327	201·9	373	367·2	419	622·1
236	47·32	282	103·6	328	204·8	374	371·8	420	628·8
237	48·20	283	105·3	329	207·6	375	376·3	421	635·6
238	49·08	284	107·0	330	210·5	376	380·9	422	642·5
239	49·98	285	108·7	331	213·4	377	385·5	423	649·4
240	50·89	286	110·4	332	216·4	378	390·2	424	656·3
241	51·83	287	112·1	333	219·4	379	394·9	425	663·3
242	52·77	288	113·9	334	222·4	380	399·6	426	670·4
243	53·72	289	115·7	335	225·4	381	404·4	427	677·5
244	54·69	290	117·5	336	228·5	382	409·3	428	684·7
245	55·68	291	119·3	337	231·6	383	414·1	429	691·9
246	56·68	292	121·2	338	234·7	384	419·0	430	699·2
247	57·69	293	123·0	339	237·8	385	424·0	431	706·5
248	58·71	294	124·9	340	241·1	386	429·0	432	713·9
249	59·75	295	126·8	341	244·3	387	434·1	433	721·4
250	60·81	296	128·8	342	247·6	388	439·2	434	728·9
251	61·89	297	130·7	343	250·9	389	444·4	435	736·5
252	62·98	298	132·7	344	254·3	390	449·6	436	744·1
253	64·09	299	134·7	345	257·6	391	454·9	437	751·8
254	65·21	300	136·8	346	261·1	392	460·2	438	759·6
255	66·35	301	138·9	347	264·5	393	465·5	439	767·4
256	67·50	302	141·0	348	268·0	394	470·9	440	775·3
257	68·66	303	143·1	349	271·5	395	476·4	441	783·2
258	69·85	304	145·2	350	275·0	396	481·9	442	791·2
259	71·05	305	147·4	351	278·6	397	487·4	443	799·3
260	72·27	306	149·6	352	282·3	398	493·0	444	807·4
261	73·50	307	151·8	353	285·9	399	498·7	445	815·6
262	74·76	308	154·0	354	289·6	400	504·4	446	823·9
263	76·03	309	156·3	355	293·4	401	510·1	447	832·2
264	77·31	310	158·6	356	297·1	402	515·9	448	840·6
265	78·62	311	160·9	357	300·9	403	521·7	449	849·0

been made up by the pressure of the vapour in A, and is equal in this case to 1 inch of mercury. The experiments of Regnault give the elasticities or pressure of vapour of water at different temperatures as in Table 67 and in column 4 of Table 68; thus at 212° the boiling point of water, column C, would have been raised to E, at the same level as B, showing that at that temperature the elastic force of vapour is equal to that of the atmosphere.

(177.) The weight of water contained in the vapour may now be calculated; the experiments of Regnault and Despretz have shown that the weight of a given volume of vapour of water is ·623, or nearly $\frac{5}{8}$ the weight of the same volume of air at the same temperature and pressure. (See Table 39.) Thus at 132°, the weight of a cubic foot of dry air at atmospheric pressure is given by column 3 of Table 68 at ·0671 lb., but the force of vapour at 132° is 4·752 inches of mercury, under which pressure air would weigh $\cdot 0671 \times 4 \cdot 752 \div 29 \cdot 921 = \cdot 010656$ lb. only, therefore a cubic foot of vapour will be $\cdot 010656 \times \cdot 623 = \cdot 006639$ lb., and thus has been calculated column 7 of Table 68.

(178.) "*Mixtures of Vapour and Air.*"—By opening the cock at F we may admit the atmospheric air; if the space A, Fig. 57, had been a vacuum we should of course have then a cubic foot of dry air there, whose weight is given by column 3 in Table 68, but vapour being present, less than a cubic foot will be required. Say we take the case in which the elastic force of the vapour is 15 inches, or about half that of the atmosphere. If we assume that the relative volumes and densities of vapour follow the law of Marriotte (29) as dry air and gases do (which is not a fact, but the correctness of our deduction will not be affected in this case), we shall have a cubic foot of vapour at 15 inches pressure or elastic force, and if a piston were fitted into the cube, as at Fig. 58, and forced to descend from A to B, the pressure would increase from 15 inches at A to 30 inches at B, and the cubic foot of vapour would of course be reduced to half a cubic foot, its density being doubled and its elastic force increased to that of the atmosphere, also the space C above the piston would be a vacuum; if now the cock F were opened, the air would fill the vacant half cubic foot, and the piston being

TABLE 68.—Of the WEIGHT of AIR, VAPOUR of WATER, and MIXTURES of AIR saturated with VAPOUR, at different Temperatures, under the ordinary Atmospheric Pressure of 29·921 inches in the Barometer.

Temp. Fahr.	Volume of Dry Air at different Temperatures, the Volume at 32° being 1·000.	Weight of a Cubic Foot of Dry Air at different Temperatures, in Pounds.	Elastic Force of Vapour in Inches of Mercury. Regnault.	Mixture of Air saturated with Vapour. Elastic Force of the Air in the Mixture of Air and Vapour in Inches of Mercury.	Mixture of Air saturated with Vapour. Weight of a Cubic Foot of the Mixture of Air and Vapour. Weight of the Air in Pounds.	Mixture of Air saturated with Vapour. Weight of a Cubic Foot of the Mixture of Air and Vapour. Weight of the Vapour in Pounds.	Mixture of Air saturated with Vapour. Weight of a Cubic Foot of the Mixture of Air and Vapour. Total Weight in Pounds.	Mixture of Air saturated with Vapour. Weight of Vapour mixed with 1 lb. of Air, in Pounds.	Mixture of Air saturated with Vapour. Weight of Dry Air mixed with 1 lb. of Vapour, in Pounds.	Cubic Feet of Vapour from 1 lb. of Water at its own Pressure in Column 4.
° 32	1·000	·0807	·181	29·740	·0802	·000304	·080504	·00379	263·81	3289
42	1·020	·0791	·267	29·654	·0784	·000440	·078840	·00561	178·18	2252
52	1·041	·0776	·388	29·533	·0766	·000627	·077227	·00819	122·17	1595
62	1·061	·0761	·556	29·365	·0747	·000881	·075581	·01179	84·79	1135
72	1·082	·0747	·785	29·136	·0727	·001221	·073921	·01680	59·54	819
82	1·102	·0733	1·092	28·829	·0706	·001667	·072267	·02361	42·35	600
92	1·122	·0720	1·501	28·420	·0684	·002250	·070717	·03289	30·40	444
102	1·143	·0707	2·036	27·885	·0659	·002997	·068897	·04547	21·98	334
112	1·163	·0694	2·731	27·190	·0631	·003946	·067046	·06253	15·99	253
122	1·184	·0682	3·621	26·300	·0599	·005142	·065042	·08584	11·65	194
132	1·204	·0671	4·752	25·169	·0564	·006639	·063039	·11771	8·49	151
142	1·224	·0660	6·165	23·756	·0524	·008473	·060873	·16170	6·18	118
152	1·245	·0649	7·930	21·991	·0477	·010716	·058416	·22465	4·45	93·3
162	1·265	·0638	10·099	19·822	·0423	·013415	·055715	·31713	3·15	74·5
172	1·285	·0628	12·758	17·163	·0360	·016682	·052682	·46338	2·16	59·2
182	1·306	·0618	15·960	13·961	·0288	·020536	·049336	·71300	1·402	48·6
192	1·326	·0609	19·828	10·093	·0205	·025142	·045642	1·22643	·815	39·8
202	1·347	·0600	24·450	5·471	·0109	·030545	·041445	2·80230	·357	32·7
212	1·367	·0591	29·921	0·000	·0000	·036820	·036820	infinite	·000	27·1
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)

removed the two would intermix with one another, but the relative weights would not be altered, the vapour would return to its normal volume and pressure of 15 inches of mercury, and the air being dilated to double its normal volume and reduced to half its normal density and pressure, or to 15 inches of mercury also.

In all cases the *sum* of the elastic forces of the vapour and air will be equal to the atmospheric pressure, the vapour taking always the force due to its temperature as per column 4 in Table 68, and the air making up the complement, as in col. 5.

(179.) Returning to our former illustration:—at 132° the vapour had a force of 4·752 inches, the pressure of the air in that case therefore must be $29 \cdot 921 - 4 \cdot 752 = 25 \cdot 169$ inches, and a cubic foot of air at that density would weigh $\cdot 0671 \times 25 \cdot 169 \div 29 \cdot 921 = \cdot 0564$ lb., and thus column 6 in Table 68 has been calculated. The total weight will of course be the sum of the separate weights or $\cdot 006639 + \cdot 0564 = \cdot 063039$ lb., so that a pound of saturated air will contain $\cdot 006639 \div \cdot 0564 = \cdot 11771$ lb. of vapour, and a pound of vapour will saturate $\cdot 0564 \div \cdot 006639 = 8 \cdot 49$ lbs. of dry air, and so of the rest as per columns 8, 9, and 10, in Table 68.

(180.) We assumed in (178), for the purpose of illustration, that vapour was subject to the law of Marriotte, and could be reduced to half its normal volume by doubling the pressure; but, in fact, the elastic force of vapour can never exceed that due to its temperature (55) as given by Table 67, and the effect of compressing it is not to increase its pressure, but to cause the surplus vapour to be condensed, that which remains retaining the pressure or elastic force due to it. The effect of mixing vapour with dry air is shown by (79).

(181.) "*Vapour in the Atmosphere.*"—The amount of vapour in the atmosphere varies exceedingly with the climate and the seasons. The air in nature is never perfectly dry: with the greatest dryness on record in this country the air still retained 23 per cent. of the vapour that would saturate it. In winter the air is frequently quite saturated with moisture: when it contains more than 85 per cent. we consider it damp; 65 per cent., moderately dry; 50 per cent., dry; 35 per cent., very dry; and 25 per cent., extremely dry.

Table 69 gives the mean humidity of the air throughout the year at Greenwich, Halle in Germany, and Madrid; these last may be taken as typical of central and southern Europe respectively. This table shows that in this country June is the dryest month of the year, holding at mid-day when the air is the dryest 58 per cent. of the moisture that would saturate it, &c. In summer the hour of greatest dryness is between 2 and 3 P.M., and in winter between 1 and 2 P.M. The hour of greatest dampness is in summer between 4 and 5 A.M., and in winter between 6 and 7 A.M. The table also shows that in this country, the mean dryness of the whole year, taken day and night, is 82 per cent., that of the *day* being 76 per cent. and of mid-day 72 per cent. of the vapour that would saturate the air.

TABLE 69.—Of the HUMIDITY of the AIR.

Month.	Greenwich.			Halle.	Madrid.
	Mean of the 24 hours.	Mean of the Day. Between 9 A.M. and 6 P.M.	Mean of the Day. Between 2 and 3 P.M.	Mean of the 24 hours.	Mean of the 24 hours.
	Mean Humidity: Saturation = 100.				
January	89	86	84	86	84
February	85	85	81	81	72
March	82	79	75	77	66
April	79	68	64	71	57
May	76	71	66	69	..
June	74	62	58	71	56
July	76	67	63	68	44
August	77	70	66	66	45
September ..	81	76	68	73	56
October	87	81	76	79	80
November	89	86	85	86	78
December	89	86	84	87	85
Year	82	76	72	76	66

(182.) "*Hygrometry.*"—The best and most convenient mode of estimating the dryness of the air is by the dry and wet-bulb hygrometer, which consists of a pair of thermometers, one of which has its bulb covered with muslin kept continually wet by a filament of thread leading to a small vessel of water, and

acting by capillary attraction; the other having its bulb dry in the usual way. The effect of the evaporation is to lower the temperature (193), and the amount of cold produced, depending on the amount of vapour present in the ambient air, indicates the dryness of the air as shown by Table 70. With a moderate temperature of the air, say 62°, 2° of cold or less indicates that the air is damp; 6° moderate dryness; 10° would show that the air is dry; 15° very dry; and 20°, extremely dry. In an inhabited room from 6° to 8° of cold, or of difference of the dry and wet bulbs will indicate a pleasant and healthful degree of humidity. The philosophy of this method of finding the dryness of the air is shown in (193).

TABLE 70.—Of the INDICATIONS of the HYGROMETER (Dry and Wet Bulb), from Mr. GLAISHER'S Observations at Greenwich.

Temperature of the Air.	Degrees of Cold in the Wet-bulb Thermometer.											
	1	2	3	4	5	6	7	8	9	10	11	12
	Degrees of Humidity, Saturation being 100.											
32°	87	75										
42	92	85	78	72	66	60	54	49	44	40	36	33
52	93	86	80	74	69	64	59	54	50	46	42	39
62	94	88	82	77	72	67	62	58	54	50	47	44
72	94	89	84	79	74	69	65	61	57	54	51	48
82	95	90	85	80	76	72	68	64	60	57	54	51
92	95	90	85	81	77	73	70	66	62	59	56	53

Temperature of the Air.	Degrees of Cold in the Wet-bulb Thermometer.											
	13	14	15	16	17	18	19	20	21	22	23	24
	Degrees of Humidity, Saturation being 100.											
42°	30	27										
52	36	33	30	27	25							
62	41	38	35	32	30	28	26	24				
72	45	42	39	36	34	32	30	28	26	24	23	22
82	48	45	42	40	38	35	33	31	29	27	26	25
92	50	47	45	43	41	38	36	34	32	30	28	26

(183.) "*Steam.*"—When the elastic force of vapour of water exceeds the pressure of the atmosphere it receives usually the name of "steam," and becomes one of the most useful mechanical agents that we have. Table 71 gives the elastic force of

TABLE 71.—Of the TEMPERATURE and VOLUME of STEAM at different PRESSURES, calculated from the Experiments of REGNAULT, &c.

Pressure above the Atmosphere in Lbs. per Sq. Inch.	Temp. Fahr.	Cubic Feet of Steam from One Cubic Foot of Water.	Pressure above the Atmosphere in Lbs. per Sq. Inch.	Temp. Fahr.	Cubic Feet of Steam from One Cubic Foot of Water.	Pressure above the Atmosphere in Lbs. per Sq. Inch.	Temp. Fahr.	Cubic Feet of Steam from One Cubic Foot of Water.
0	212°	1640	44	291°	446	88	330°	264
1	215	1544	45	292	439	89	”	261
2	219	1456	46	294	432	90	331	258
3	222	1378	47	295	426	91	332	256
4	224	1310	48	296	419	92	333	254
5	227	1247	49	297	413	93	”	252
6	230	1190	50	298	407	94	334	250
7	232	1138	51	299	401	95	”	248
8	235	1089	52	300	394	96	335	246
9	237	1048	53	301	387	97	336	244
10	239	1008	54	302	384	98	”	242
11	241	969	55	303	379	99	337	240
12	244	936	56	304	374	100	338	238
13	246	904	57	305	369	102	339	234
14	248	875	58	”	364	104	340	230
15	250	846	59	306	360	105	341	228
16	252	820	60	307	355	106	342	226
17	253	796	61	308	350	108	343	223
18	255	774	62	309	346	110	344	220
19	257	743	63	310	342	115	347	212
20	259	732	64	311	338	120	350	203
21	260	713	65	312	334	125	353	197
22	262	694	66	313	330	130	355	190
23	264	676	67	314	326	135	358	184
24	265	660	68	”	322	140	361	179
25	267	644	69	315	319	145	363	174
26	268	630	70	316	315	150	366	169
27	270	615	71	317	312	160	370	159
28	271	602	72	318	309	170	375	151
29	272	589	73	”	306	180	380	144
30	274	576	74	319	302	190	384	138
31	275	564	75	320	299	200	388	132
32	277	553	76	321	296	210	392	126
33	278	542	77	322	293	220	396	121
34	279	532	78	”	290	230	399	116
35	281	521	79	323	287	240	403	112
36	282	512	80	324	284	250	406	108
37	283	503	81	325	282	260	409	104
38	284	494	82	”	279	270	413	101
39	285	485	83	326	276	280	416	98
40	286	476	84	327	273	290	419	95
41	288	468	85	328	271	300	422	91
42	289	460	86	”	268			
43	290	453	87	329	266			

steam at different temperatures, not in inches of mercury above a vacuum as in Table 67, but in pounds per square inch above atmosphere, in order to adapt it to practice, and the volume of steam per cubic foot of water at different pressures is given for the same purpose.

(184.) "*The Elastic Force of Vapour of Alcohol, &c.*"—We have so far considered only the force of vapour of water; that of other liquids is very different, as shown by Table 72; thus while at 212° the force of vapour of water is just equal to that of the atmosphere, alcohol gives 2·2 atmospheres, ether 6·5 atmospheres above vacuum, &c., &c.

TABLE 72.—Of the ELASTIC FORCE of the VAPOUR of ALCOHOL, ETHER, &c., &c., in Inches of Mercury, from the Experiments of REGNAULT.

Temp. Fahr.	Water.	Alcohol.	Ether.	Oil of Turpentine.	Sulphuret of Carbon.	Chloroform.
° −4	·036	·130	2·724	..	..	..
+14	·082	·256	4·457	..	3·110	..
32	·181	·500	7·177	·082	5·012	..
50	·361	·949	10·100	·091	7·846	5·133
68	·685	1·732	17·118	·169	11·740	7·488
86	1·242	3·087	25·079	·276	17·110	10·870
104	2·162	5·279	35·968	·441	24·311	14·331
122	3·621	8·673	49·921	·677	33·571	20·642
140	5·858	13·780	68·122	1·059	45·772	29·055
158	9·177	21·228	90·925	1·649	60·982	38·433
176	13·962	32·000	118·600	2·409	88·941	53·850
194	20·687	47·866	153·504	3·583	112·035	71·319
212	29·921	66·338	193·717	5·311	130·760	92·701
230	42·337	92·591	246·024	7·374	162·846	118·913
248	58·712	126·291	..	10·118	201·638	150·315
266	79·932	170·520	..	13·661	246·480	185·866
284	107·000	221·957	..	18·201	..	..
302	141·000	285·740	..	23·799	..	..

The remarkable law of Dalton, explained in (14), as applied to the boiling points of liquids, applies equally to the elastic forces of the vapours of those liquids, but the experiments of Regnault show that it is only approximately correct. The elastic force of ether at any particular temperature should by Dalton's law be equal to that of water at 212 − 100 = 112°

higher temperature, oil of turpentine should be equal to water 316 − 212 = 104° lower, &c., &c.

Thus the elastic force of ether, say at 140°, should be equal to that of water at 140 + 112 = 252°, or 62·98 inches by Table 67; Regnault's experiments in Table 72 give 68·122 inches. Again, alcohol at say 230° should be equal to water at 230 + (212 − 173) = 269°, or 84·01 inches by Table 67, whereas Table 72 gives 92·591 inches. Oil of turpentine at 230° should be equal to water at 230 − (316 − 212) = 126°, or 4·039 inches by Table 67, but Table 72 gives 7·374 inches, &c.

CHAPTER VII.

ON EVAPORATION.

(185.) "*Evaporation in Open Air.*"—When a liquid such as water is freely exposed to the atmosphere, the stratum of air in contact with its surface becomes more or less charged with vapour, and instead of becoming heavier thereby, as might be supposed, its specific gravity is less than before. This is shown by comparing cols. 3 and 8 of Table 68, and it will be observed that the difference increases rapidly with increase of temperature: thus at 52°, saturated air is ·077227 ÷ ·0776 = ·99, or 99 per cent. of the weight of the same volume of dry air, but at 202° it is only ·041445 ÷ ·06 = ·69, or 69 per cent. The result is that the moist air rises and is replaced by dry air, which in its turn becomes similarly charged and bears off its load of water.

If the air were saturated with vapour to begin with, it could obviously carry no more, and evaporation would cease; but if it were only partially saturated, the process would proceed, only more slowly than with dry air. As we have shown in (181), the atmosphere in nature is always more or less charged with vapour, the mean for the whole year in this country being about 82 per cent. of the vapour that would saturate it.

(186.) The rate of evaporation with different liquids and at different temperatures is proportional to the *difference* of the elastic forces of the vapour of the liquid at the given temperature of that liquid, and that of the vapour actually present in the ambient air.

To obtain data on this subject, I made a series of experiments on the rate of evaporation of water, alcohol, benzoline, and ether, by exposing them to perfectly calm air at natural temperatures, in a vessel suspended by a delicate balance, and thus obtained the following results.

With water, 30 grains were evaporated in 620 minutes, or $30 \times 60 \div 620 = 2 \cdot 9$ grains per hour from a vessel $3\frac{1}{16}$ inches diameter $= 7 \cdot 366$ square inches area, which is equal to $2 \cdot 9 \times 144 \div 7 \cdot 366 = 56 \cdot 7$ grains per square foot per hour. The mean temperature of the ambient air was 60°, and the wet-bulb hygrometer 55°, showing humidity 71 per cent. of saturation.

With common alcohol 56 overproof, and air at 60°, the same vessel evaporated 120 grains in 6 hours, or 20 grains per hour, which is equal to $20 \times 144 \div 7 \cdot 366 = 391$ grains per square foot per hour.

With benzoline, 90 grains were evaporated in 90 minutes, or 60 grains per hour, equivalent to $60 \times 144 \div 7 \cdot 366 = 1173$ grains per square foot per hour.

With ether, specific gravity ·735 and air at 57°, the evaporating vessel was 1 inch square, and 33 grains were evaporated in an hour, or $33 \times 144 = 4752$ grains per square foot per hour.

We have thus for these four liquids 56·7, 391, 1173, and 4752 grains respectively; the ratios are 1, 6·9, 20·69, and 83·81, or say 1, 7, 21, and 84 respectively.

These experimental results agree with the rule :—

$$R = (F - f) \times 378,$$

in which $R =$ the rate of evaporation in grains per square foot per hour with the air perfectly calm, $F =$ the force of the vapour of the given liquid at the temperature of the ambient air, as distinguished from that of the liquid itself (186), which may be lowered by evaporation (193); and $f =$ the force

of the vapour of water actually present in the air as indicated by the hygrometer.

In our experiments on water, the air was at 60°, and by Table 67, F = ·518 inch, therefore with the wet-bulb at 55° showing humidity ·71, f = ·518 × ·71 = ·368 inch, and by the rule we have (·518 – ·368) × 378 = 56·7 grains per square foot per hour, or the same as by experiment.

With alcohol, by interpolation with Table 72, the force of vapour of alcohol at 60° or F is 1·384 inch, and f being ·368 inch as before, the rule gives (1·384 – ·368) × 378 = 384 grains per square foot per hour: experiment gave 391 grains.

With ether, the air was at 57°, and the wet-bulb 53° showing humidity ·75, the force of vapour of ether at 57° or F is 12·83 inches by Table 72, and f being ·465 × ·75 = ·348, the rule becomes (12·83 – ·348) × 378 = 4718 grains per square foot per hour: experiment gave 4752 grains.

Thus by the rule the ratios are 1, 6·77, and 83·21, and by experiment 1, 6·9, and 83·81 respectively.

(187.) With the very volatile fluids ether and benzoline the rate of evaporation at first before the *régime* is established, is considerably greater than the mean rate we have given. Thus with ether, the first successive weights of 1 grain were evaporated from the vessel 1 inch square in 77, 87, and 95 seconds respectively, which were equivalent to 6725, 5962, and 5457 grains per square foot per hour. Similarly with benzoline the first successive weights of 5 grains were evaporated from the vessel $3\frac{1}{16}$ inches diameter in 215, 238, 264, 278, 286, and 300 seconds respectively, or 1642, 1484, 1338, 1270, 1235, and 1177 grains per square foot per hour.

(188.) The rate of evaporation of water and other liquids whose elastic force of vapour is low varies very much with the humidity of the ambient air, while that of ether, &c., is little affected, as may be seen by studying the calculations (186). Table 73, which has been calculated by the rule in (186), shows the effect of different degrees of humidity on the evaporation of water at natural temperatures.

(189.) "*Effect of Wind on Evaporation.*"—The rules and experiments we have so far given apply only to perfectly calm

TABLE 73.—Of EVAPORATION AT NATURAL TEMPERATURES, and with Air in different states of Dryness.

Temp. of the Air and Water.	Humidity of the Air: Saturation = 100.							
	Dry	30	40	50	60	70	80	90
	Grains evaporated per Square Foot per Hour in calm Air.							
32	69	48	41	34	28	21	14	7
42	101	71	61	51	40	30	20	10
52	147	103	88	74	59	44	29	15
62	211	148	127	106	84	63	42	21
72	298	209	178	149	119	89	60	30
82	426	298	256	213	170	128	85	43
92	570	400	342	285	228	171	114	57

air when the necessary motion of the air ensues only from the change of density due to the presence of vapour, as we have seen in (185), the air charged more or less with moisture ascending vertically. If that motion be prevented by a cover, even a loosely fitting one, the confined air becomes fully saturated, and not being able to get away with its load, evaporation ceases, and in any case, even if the air be allowed to rise, the motion is very slow, and is retarded by the dry air descending to replace it. With a wind, the stratum of air in contact with the water moves off horizontally, not waiting to be fully saturated, but taking only a small percentage of that amount. When air moderately dry comes in contact with a surface of water, it absorbs moisture at first with avidity, but as it becomes saturated the process proceeds more and more slowly, and finally ceases; evidently, therefore, the more rapidly the air is renewed, the more rapidly the evaporation proceeds.

(190.) To ascertain the effect of wind on evaporation, the vessel used in the former experiments, which was $3\frac{1}{16}$ inches diameter and $\frac{3}{4}$ inch deep, was about half filled with water, and exposed to westerly winds, which conveniently varied in strength on three successive days from a fresh breeze to a gale.

With a fresh breeze 17, 34, and 63 grains were evaporated in 1, 2, and 4 hours respectively: taking 17 grains per hour as a mean, we have $17 \times 144 \div 7 \cdot 366 = 332$ grains per square foot per hour. The air was at 57° and the wet-bulb at 49°,

indicating $\cdot 57$ humidity; hence $F = \cdot 465$, $f = \cdot 465 \times \cdot 57 = \cdot 265$, and with calm air we should have had by the rule $(\cdot 465 - \cdot 265) \times 378 = 76$ grains: the ratio with a fresh breeze is $332 \div 76 = 4 \cdot 4$ to 1 with calm air.

With a strong wind 28 grains per hour, or $28 \times 144 \div 7 \cdot 366 = 547$ grains per square foot per hour were evaporated; the air was at 53°, the wet-bulb 46°, humidity $\cdot 59$; hence $F = \cdot 403$ and $f = \cdot 403 \times \cdot 59 = \cdot 238$, and with calm air we should have $(\cdot 403 - \cdot 238) \times 378 = 62$ grains: the ratio is $547 \div 62 = 8 \cdot 8$ to 1.

With a gale 24 grains per hour, or $24 \times 144 \div 7 \cdot 366 = 470$ grains per square foot per hour were evaporated, which though actually less than in the last experiment, was *relatively* greater, as we shall see; the air was at 52°, the wet-bulb 48°, humidity $\cdot 74$, hence $F = \cdot 388$, and $f = \cdot 388 \times \cdot 74 = \cdot 287$, and the rule gives with calm air $(\cdot 388 - \cdot 287) \times 378 = 38$ grains per square foot per hour: the ratio is $470 \div 38 = 12 \cdot 4$ to 1.

Thus we have the ratios 1, 4·4, 8·8, and 12·4 with calm air, a fresh breeze, a strong wind, and a gale respectively.

(191.) Similar results were obtained with alcohol which was exposed in the vessel 1 inch square to the same gale as in the last experiment on water; $22\frac{1}{2}$ grains per hour, or $22 \cdot 5 \times 144 = 3240$ grains per square foot were evaporated. The air was at 52°, the wet-bulb 48°, or humidity $\cdot 74$, and by Table 72 $F = 1 \cdot 036$, $f = \cdot 388 \times \cdot 74 = \cdot 287$ by Table 67, and in calm air we should have had $(1 \cdot 036 - \cdot 287) \times 378 = 283$ grains: the ratio is $3240 \div 283 = 11 \cdot 45$ to 1, or nearly the same as water, which was 12·4 with the same wind, &c.

To confirm this result, and to show that the smallness of the vessel had no effect on the correctness of the experiment, it was exposed to *calm* air, and successive weights of 1 grain were evaporated in 36, 35, 34, 36, 34 minutes. Taking 35 minutes as a mean, we have $144 \times 60 \div 35 = 247$ grains per square foot per hour. The air was at 50°, the wet-bulb 47°, indicating humidity $\cdot 8$; then $F = \cdot 949$ and $f = \cdot 361 \times \cdot 8 = \cdot 289$, and by the rule with calm air we have $(\cdot 949 - \cdot 289) \times 378 = 250$ grains per square foot per hour, or almost precisely the same as by experiment.

TABLE 74.—Of EXPERIMENTS on the RATE of EVAPORATION of different LIQUIDS at NATURAL TEMPERATURES; showing also the effect of Wind on the result.

Liquid.	State of the Wind.	Size of Vessel.	Grains Evaporated per Hour.	Grains Evaporated per Square Foot per Hour by		Ratio with Wind.	Temp. of the Air.	Temp. of Wet-bulb.	Humidity. Saturation = 1·0.	F.	f.
				Expt.	Rule, for Calm Air.						
							°	°			
Water ..	Dead Calm ..	$3\frac{1}{16}$	2·9	56·7	56·7	..	60	55	·71	·518	·368
Alcohol ..	Do.	$3\frac{1}{16}$	20	391	384	..	60	55	·71	1·384	·368
Ether ..	Do.	1 sq	33	4752	4718	..	57	53	·75	12·83	·348
Water ..	Fresh Breeze	$3\frac{1}{16}$	17	332	81	4·4	57	49	·57	·465	·265
Do. ..	Strong Wind	$3\frac{1}{16}$	28	547	62	8·8	53	46	·59	·403	·238
Do. ..	Gale	$3\frac{1}{16}$	24	470	38	12·4	52	48	·74	·388	·287
Alcohol ..	Gale	1 sq	$22\frac{1}{2}$	3240	285	11·45	52	48	·74	1·036	·287
Do. ..	Dead Calm ..	1 sq	1·715	247	250	..	50	47	·80	·949	·289
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)

Table 74 gives a collective statement of the results of the experiments and calculations.

(192.) The observations of meteorologists on the amount of evaporation in nature are very discordant, the differences probably arising from position of the gauge as to exposure to wind, &c., which, as we have seen, is very influential on the result. Taking the question to be the evaporation from a large surface of water such as a river, the observations of Mr. Greaves seem the most reliable, being made with a light slate cistern set afloat by a raft in a mill-stream. It was 3 feet square, 1 foot deep, was immersed to a depth of 4 inches, had 4 to 6 inches of water continually in it, being never emptied; the depth was taken with a dip-stick, and being compared with an adjacent rain-gauge, gave the evaporation alone. This cistern was fully exposed to the sun, wind, rain, and evaporation: the mean result of ten years' observations, 1860 to 1869 inclusive, was 21 inches depth evaporated per annum, the mean rainfall for the same interval being 25½ inches. But Mr. Fletcher found more than double this quantity, or 47½ inches, as the mean of seven years' observation with a small vessel fully exposed, 5 feet above the ground; and Mr. Miller with a similar apparatus obtained 30 inches, as shown by Table 75, which is useful as giving the proportions for the several months of the year.

TABLE 75.—Of the EVAPORATION of WATER at NATURAL TEMPERATURES, from observations at Whitehaven by Mr. MILLER.

	Inches.		Inches.
January	·880	August	3·398
February	1·042	September	3·174
March	1·770	October	1·930
April	2·535	November	1·322
May	4·146	December	1·087
June	4·547		
July	4·200	Total	30·032

NOTE.—The above is the mean of six years, during which the mean rainfall was 45·255 inches.

(193.) "*Cold produced by Evaporation.*"—It is shown by (17) that when water changes its state from the liquid to the vaporous, a large amount of heat is absorbed by the vapour

and becomes latent; thus to vapourize a pound of water at 62° requires by the common rule 1178 − 62 = 1116 units of heat. If we assume the temperature of the water to be constant, the heat thus required must be derived from the air in contact with the surface, which air will be cooled to an extent varying with the volume used. Taking water and air both at 62°, and assuming that the air is perfectly dry on arrival and perfectly saturated on departure, then by col. 10 of Table 68, each pound of water requires 84·79 lbs. of *dry* air, which having to yield, as we have seen, 1116 units of heat, must be cooled 1116 ÷ (84·79 × ·238) = 55°·3, and the film of water in contact with it would also be cooled to the same extent.

But to produce this result, the film of surface water must not receive heat at the same time from the mass of the liquid or from other sources. Taking for illustration the wet-bulb thermometer (182), as soon as depression commences the bulb begins to receive heat from the ambient air, the amount increasing with the depression until it is equal to that given out by evaporation, and the lowering of temperature is arrested at a point much higher than that due to evaporation alone. If the depression is thus reduced to half or 55·3 ÷ 2 = 27°·65, then as the air is cooled only half, we must have double the volume, &c.

Experience alone can determine at what point the heat given out by evaporation is equal to that received by contact of air. Table 76 gives the results of Gay-Lussac's experiments on this subject, which were made by causing a current of air previously dried by chloride of calcium (229) to pass over the moistened bulb of a thermometer. For 62°, this table gives a depression of 20°·7 only, instead of 55°·3, from which it would appear that the air departs only 20·7 ÷ 55·3 = ·38, or 38 per cent. saturated; hence 84·79 ÷ ·38 = 223 lbs. of dry air per pound of water were used. If the air had been half saturated to begin with, instead of being dry, we should have required a double quantity, or 446 lbs. of air per pound of water, which would be cooled 1116 ÷ (446 × ·238) = 10°·5, which agrees with Table 70, namely, that air at 62° and 10° of cold, indicates air half saturated, or 50 per cent.

Table 76 shows that the ratio increases with the difference between the air and the wet-bulb, as might be expected; with 10°·4 it is ·57, and with 28°·6 it is ·262, &c.

In the open air in this country, the average maximum depression is 2° in January, rising gradually to 9° in June, thence decreasing to the end of the year. Occasionally a great degree of cold is obtained; Mr. Arnold observed as much as 25° in June at Aldershott, the dry and wet bulbs showing 82° and 57° respectively. In India Colonel Sykes observed 33° of cold, or 98° and 65° respectively.

This simple method of producing cold is extensively used in India and other tropical climates for cooling water, wine, &c.; a thick layer of straw is spread on the ground in the shade of a building, but exposed as much as possible to the wind, the bottles to be cooled are placed on it, more straw is lightly spread over them, and the whole is kept wet by frequent watering. With the air at 98°, the bottles will cool in a wind to 65°, but in calm air to 71° only.

TABLE 76.—Of the COLD produced by EVAPORATION of WATER with AIR perfectly dry; calculated from GAY-LUSSAC's experiments.

Temp. of Dry Air.	Temp. of Wet-bulb.	Cold produced.		Ratio.
		By Experiment.	Maximum calculated.	
°	°	°	°	
32	21·6	10 4	18·5	·57
42	28·6	13·4	26·8	·50
52	35·1	16·9	38·8	·44
62	41·3	20·7	55·3	·38
72	47·4	24·6	71·1	·346
82	53·4	28·6	109·0	·262

(194.) "*Cold by Evaporation of Ether, &c.*"—The degree of cold produced by the evaporation of very volatile liquids, such as ether, alcohol, &c., is much greater than that due to water. Thus with air at 60°, and 66 per cent. saturated with vapour, the cold produced by water is 6° by Table 70, but that with ether was found by experiment to be 42°, or seven times greater than water, the temperature being reduced to 60 − 42 = 18°

But here experiment gave a remarkable result; the depression was not permanent, although the wet-bulb was fed continuously with ether by threads acting by capillary attraction, the depression diminished rapidly: being seven times that of water, or 42° at first, it became only six times, or 36° in 5 minutes; five times, or 30° in 11 minutes; four times, or 24° in 17 minutes; three times, or 18° in 27 minutes; and double only, or 12° in 60 minutes.

Alcohol gave similar results; at first the depression was 9°, or 3° greater than water, but became only $2\frac{1}{4}$° in 19 minutes; $1\frac{1}{2}$° in 70 minutes; 1° in 107 minutes; $\frac{3}{4}$° in 142 minutes; $\frac{1}{2}$° in 165 minutes, and 0° in 290 minutes. The explanation probably is that the *pure* spirit rapidly evaporates; leaving on the wet-bulb a *diluted* spirit approximating more and more nearly to water in the course of time.

With benzoline the result was still more remarkable: at first the depression was 10°, or double that of water (which was 5° at that time), but became only 9° in 5 minutes, 7° in 10 minutes, 4° in 35 minutes, 3° in 85 minutes, and $2\frac{1}{4}$° in 135 minutes. With reference to water, the depression was at first double, became equal in 25 minutes, and only half in 2 hours.

(195.) "*Evaporation at High Temperatures, but below Boiling Point.*"—We have so far considered the case in which the water and air had one and the same temperature: we will now investigate the phenomena of evaporation from open vessels heated by a fire beneath, &c., to a temperature superior to that of the ambient air, but below that of ebullition.

There are two important questions to be determined, namely, the heat required to evaporate water at different temperatures, and the rate of evaporation. To obtain these data I made the two sets of experiments, the results of which are given in Table 77, by suspending from a very delicate balance a vessel 12 inches square, $2\frac{1}{4}$ inches deep, containing about $7\frac{3}{4}$ lbs. of hot water. To prevent loss of heat laterally, this vessel was enclosed in another of larger dimensions, the interval beneath and at the sides being filled with about an inch of wadding. In a certain observed *time*, the water lost a certain *weight* by evaporation, and a certain amount of *heat* by which that evapora-

TABLE 77.—Of EXPERIMENTS on the EVAPORATION of WATER at different TEMPERATURES, from open Vessels exposed to Air at 52°, and Humidity 86.

Weight of Water.	Temperatures as observed.	Weight Evaporated.	Time to Evaporate the Weight in Col. 3.	Water Evaporated per Sq. Foot per Hour.	Mean Temperature of the Water.	Mean Weight of Water.		Loss of Heat by the Weight in Col. 8.	Units of Heat to Evaporate Col. 3 from Temperature in Col 6.	Heat to Evaporate 1 lb. of Water from The Temp. in Col. 6.	Heat to Evaporate 1 lb. of Water from 32°.
grains	°	grains	seconds	grains	°	grains	lbs.	°		units	units
54280	190	480	110	15709	185½	54040	7·720	9	69·48	1013	1167
53800	181	480	145	11920	175¾	53560	7·651	10⅓	80·33	1171	1315
53320	170½	480	190	9095	164¾	53080	7 583	11½	87·20	1272	1399
52840	159	480	270	6400	153½	52600	7·514	11½	86·41	1260	1382
52360	147½	480	395	4375	141½	52120	7·446	12⅓	93·07	1357	1467
51880	135	480	600	2880	128¼	51640	7·377	13½	99·59	1452	1548
51400	121½	480	1020	1694	114	51160	7·308	15¼	111·40	1624	1716
50920	106¼	480	2060	839	97	50680	7·240	18¾	135·70	1979	2044
50440	87½	180	1425	455	83½	50350	7·193	8	57·54	2238	2290
50260	79½										
SECOND EXPERIMENT.											
54280	195	480	85	20330	190	54040	7·720	10	77·20	1126	1284
53800	185	480	135	12800	179½	53560	7·651	11	84·15	1227	1375
53320	174	480	175	9874	168¼	53080	7·583	11⅓	87·20	1272	1408
52840	162½	480	235	7353	157	52600	7·514	11	82·65	1206	1331
52360	151½	480	330	5236	145½	52120	7·446	12	89·35	1303	1417
51880	139½	480	505	3422	132¾	51640	7·377	13½	99·59	1452	1553
51400	126	480	825	2095	118¾	51160	7·308	14½	107·4	1566	1653
50920	111½	480	1520	1137	102¾	50680	7·240	17½	126·7	1848	1919
50440	94	480	3960	436	83¼	50200	7·170	21½	154·2	2249	2300
49960	72½	100	1680	214	70	49910	7·130	5	35·65	2496	2534
49860	67½	120	3480	124	64¼	49800	7·114	6⅓	46·24	2697	2729
49740	61	100	5100	70	58¾	49690	7·099	4½	31·94	2236	2263
49640	56½										
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)

tion was effected, and from those facts we can obtain the data we require. Thus, taking the first experiment, 480 grains were evaporated in 110 seconds, and the temperature of the mass of water whose *mean* weight was 7·72 lbs. was reduced 9°, or from 190° to 181°. Then, the mean temperature of the water being 185½°, the rate of evaporation per hour would be $480 \times 3600 \div 110 = 15709$ grains per square foot. The water parted with $7{\cdot}72 \times 9 = 69{\cdot}48$ units of heat in vapourizing 480 grains; therefore, to evaporate 1 lb. or 7000 grains, we should require $69{\cdot}48 \times 7000 \div 480 = 1013$ units: we thus obtain cols. 5 and 11 in Table 77. By plotting these results in diagrams, and drawing mean curves we eliminate the errors of observation, and thus obtain cols. 2 and 10 in Table 79.

(196.) The rate of evaporation as shown by the experiments is represented satisfactorily by the rule:—

$$E = \{243 + (3{\cdot}7 \times t\} \times (V - v),$$

in which E = Evaporation per square foot per hour in grains,
t = Temperature of the water,
V = Force of vapour at the temperature t, from Table 67
v = Force of vapour actually present in the air

Thus, during the experiments on evaporation (195) the air was at 52° and humidity 86, then $V = {\cdot}388$, and $v = {\cdot}388 \times {\cdot}86 = {\cdot}334$, and with water and air both at 52°, $E = \{243 + (3{\cdot}7 \times 52\} \times ({\cdot}388 - {\cdot}334) = 23{\cdot}5$ grains per square foot per hour. With water, say at 152°, $V = 7{\cdot}93$, the ambient air is heated by the water from 52° to 152°, but the force of the vapour in it to begin with, or v, is still ·334 inch only, therefore $E = \{243 + (3{\cdot}7 \times 152\} \times (7{\cdot}93 - {\cdot}334) = 6115$ grains per square foot per hour. If the air had been perfectly dry, v would be 0 in both cases, and the rate of evaporation at 52° would become 169 grains, and at 152°, 6384 grains per square foot per hour, showing that humidity of the air is most influential on the result at low temperatures. Table 78 gives a combined view of the experiments, col. 5 is calculated by the rule; omitting the last experiment, which was anomalous, the

sum of col. 4 is 88,700, and of col. 5, 88,680, showing the general agreement of the rule with the experiments.

TABLE 78.—Of the COMBINED RESULTS of EXPERIMENT and CALCULATION on EVAPORATION from OPEN VESSELS.

No. of Experiment.	Mean Temperature of Water.	Heat to Evaporate 1 lb. Water. By Experiment.	Grains Evaporated per Square Foot per Hour.		Error.
			By Experiment.	By Rule.	
	°				
2	58¾	2236	70	74	− 4
2	64¼	2697	124	128	− 4
2	70	2496	214	198	+ 16
2	83¼	2249	436	442	− 6
1	83½	2238	455	449	+ 6
1	97	1979	839	853	− 14
2	102¾	1848	1137	1089	+ 48
1	114	1624	1624	1702	− 78
2	118¾	1566	2095	2257	− 162
1	128¼	1452	2880	2839	+ 41
2	132¾	1452	3422	3313	+ 109
1	141½	1357	4375	4407	− 32
2	145½	1303	5236	5237	− 1
1	153½	1260	6400	6674	− 274
1	164¾	1272	9095	8900	+ 195
2	168¼	1272	9874	9832	+ 42
1	175¾	1171	11920	12100	− 180
2	179½	1227	12800	12486	+ 314
1	185½	1013	15709	15700	+ 9
2	190	1126			
(1)	(2)	(3)	(4)	(5)	(6)

(197.) Table 79 gives a *résumé* of the phenomena in evaporating water at different temperatures, cols. 2 and 10 being based on the experiments. Thus, say we take the temperature at 122°, the air being 52° and its humidity 86. The rate of evaporation by col. 2 is 2240 grains per hour, and from this we obtain cols. 3 and 4. Then the time to evaporate 1 lb. or 7000 grains will be $7000 \div 2240 = 3 \cdot 13$ hours, as in col. 5. The total heat required to evaporate 1 lb. is 1640 units by col. 10, and this is made up of three items: 1st, the latent heat of vapourization; 2nd, loss by radiation from the surface; and 3rd, by the air necessary to combine with the vapour, which, being heated from 52° to 122°, absorbs a certain amount of heat.

The latent heat of vapourization by the correct rule in (19)

is $1115 \cdot 2 - (\cdot 708 \times 122) = 1029$ units. There is also the loss by radiation from the surface during the 3·13 hours occupied by the process: the value of R for water by Table 95 is 1·0853, and by the simple rule (276) the loss would be $1 \cdot 0853 \times (122 - 52) = 76$ units per hour; but applying the correction of Table 104 as explained in (314), we find the ratio with $122 - 52 = 70°$ excess, and 52° temperature of recipient, to be 1·2, and the true loss becomes $76 \times 1 \cdot 2 = 91 \cdot 2$ units per square foot per hour, or $91 \cdot 2 \times 3 \cdot 13 = 285$ units for the whole time, and we thus obtain cols. 6 and 7. The loss of heat by the air is more difficult to calculate, as we do not know the volume that will be used, and which has to be heated. By col. 10 of Table 68 we find that it cannot be less than 11·65 lbs. of *dry* air per pound of water, the air arriving perfectly dry and departing perfectly saturated. Neither of these conditions would be fulfilled in practice (181), (193); we may get at the volume in another way, namely, from the heat which it actually carries off. The *total* heat by col. 10 being 1640 units, and the sum of the latent heat and loss by radiation being $1029 + 285 = 1314$ units, the heat carried off by the air must be $1640 - 1314 = 326$ units, as per col. 8, and the weight of air heated from 52° to 122°, or 70°, to absorb that heat would be $326 \div (70 \times \cdot 238) = 19 \cdot 6$ lbs., from which we infer that it departs only $11 \cdot 65 \div 19 \cdot 6 = \cdot 59$, or 59 per cent. saturated. Having thus found the weight of air, the volume by Table 24 is $19 \cdot 6 \div \cdot 0776 = 253$ cubic feet, &c., as in cols. 8, 13, 14. The volume of air for evaporation at 52° has been found by inverting this process: here the air has to supply the whole of the heat necessary for effecting the vapourization, and the water and air being both at the same temperature there will be no loss by radiation. The heat that becomes latent in vapourizing a pound of water at 52° is 1078 units by col. 9. This heat has to be supplied by the air, and as by Table 70 air at 52° and humidity 86° shows 2° of cold, the weight of air which will yield 1078 units by cooling 2° is $1078 \div (2 \times \cdot 238) = 2265$ lbs., or $2265 \div \cdot 0776 = 29190$ cubic feet at 52°.

Table 79 may be applied to the solution of many practical questions, as we shall proceed to show.

TABLE 79.—Of the HEAT required to EVAPORATE ONE POUND of WATER at TEMPERATURES below the BOILING POINT, from Open Vessels exposed to Air at 52°, and Humidity 86

Temp. of the Water.	Water Evaporated per Square Foot per Hour in Calm Air. By Experiment.			Time to Evaporate 1 lb. of Water, in Hours.	Heat lost by Radiation from Surface, in Units.		Heat carried off by the Air, in Units.	Latent Heat of Vapourization. Units.	Total Heat to Evaporate 1 lb. of Water, in Units. By Experiment.		Total Heat given out per Sq. Foot per Hour, in Units.	Air at 52°, and Humidity 86, to Evaporate 1 lb. of Water.	
	Grains.	Lbs.	Depth in Inches.		Per Hour.	Total.			From the Temperature in Col. 1.	From 52°.		Lbs.	Cubic Feet.
° 52	20	·0028	·00055	350·0	0·0	0	..	1078	..	..	..	2265	29190
62	100	·0143	·00275	70·0	11·3	791	888	1071	2750	2760	39	373	4807
72	240	·0343	·0066	29·2	23·4	683	753	1064	2500	2520	86	158	2036
82	430	·0615	·0118	16·3	35·2	574	649	1057	2280	2310	140	91	1160
92	690	·0986	·0190	10·2	47·0	475	555	1050	2080	2120	204	58	747
102	1050	·150	·0288	6·67	62·7	418	449	1043	1910	1960	287	37·7	486
112	1550	·221	·0425	4·52	76·7	347	387	1036	1770	1830	392	27·1	350
122	2240	·315	·0606	3·13	91·2	285	326	1029	1640	1710	524	19·6	253
132	3180	·454	·0873	2·20	106·8	235	278	1022	1535	1615	698	14·6	184
142	4440	·634	·122	1·58	122·5	194	241	1015	1450	1540	918	11·3	146
152	6100	·871	·168	1·15	141·1	162	206	1008	1376	1476	1197	8·65	112
162	8250	1·18	·227	·848	156·4	133	193	1000	1326	1436	1564	7·37	95
172	10990	1·57	·302	·637	175·5	112	179	993	1284	1404	2016	6·27	81
182	14430	2·06	·396	·485	193·2	94	168	986	1248	1378	2573	5·43	70
192	18690	2·66	·512	·374	215·7	81	164	979	1224	1364	3268	4·92	64
202	23900	3·41	·656	·293	237·7	70	161	972	1203	1353	4106	4·51	58
212	30200	4·32	·831	·232	257·0	60	160	966	1186	1346	5112	4·20	54
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)

(198.) "*Evaporating Pans.*"—In concentrating syrups, and in many chemical operations, it is frequently essential that evaporation should be effected at low temperatures, because great heat would injure the product operated on. For such cases the evaporating pan must expose a large area and must have no cover (189), the depth is unimportant: say we had to concentrate 10 gallons of syrup to 6 gallons per hour; that is to say, 4 gallons or 40 lbs. had to be evaporated at a temperature not exceeding 152°. Table 79 shows, col. 3, that at 152° the rate of evaporation is ·871 lb. per square foot per hour; we shall therefore require 40 ÷ ·871 = 46 square feet area, say 7 ft. 8 in. diameter. The heat required to do the work is by col. 10, 1376 × 40 = 55040 units, which may be obtained by a fire beneath consuming 55040 ÷ 6000 = 9·2 lbs. of coal, allowing 6000 units as the economic value of coals for such a case (112).

(199.) "*Refrigerators.*"—In the refrigerators commonly used in breweries for cooling the wort, a very large surface is necessary to effect the purpose quickly, and we can calculate by Table 79 the area for any particular case. Say we had to cool 20 barrels of 36 gallons or 360 lbs. each, from 212° to 82° in an hour. The work to be done is to dissipate 360 × 20 × 120° = 864000 units per hour: at the commencement, each square foot will lose by col. 12, 5112 units, but at last 140 units only; taking the loss at each temperature between the extremes, we obtain an average of 1640 units per square foot per hour; hence we require 864000 ÷ 1640 = 516 square feet of surface. A large proportion of the heat dissipated is expended in evaporating a portion of the water; the mean total heat to evaporate a pound of water between 82° and 212°, by col. 10, being 1540 units, we shall vapourize 864000 ÷ 1540 = 561 lbs. The wort will therefore be concentrated by this method of cooling, losing 561 ÷ (360 × 20) = ·078, or 7·8 per cent. of the water in it.

(200.) "*Condensation Reservoirs to Steam-engines.*"—Say we take the case of an engine working day and night, and assume that the water for condensation is taken from the reservoir at 82°, the air being at 52°. Admitting that a cubic foot of water

evaporated is equal to one nominal horse-power (118), and that the temperature of that water taken from the hot-well of the air-pump is 122°, we shall have (1178 − 122) × 62·3 = 65790 units of heat per horse-power per hour to consume or dissipate. The water enters one end of the reservoir at 122°, and departs to the engine at the other end at 82°, being gradually cooled 40° in its passage. By col. 12 of Table 79 the mean rate of loss of heat between 82° and 122° is 310 units per hour: we shall therefore require 65790 ÷ 310 = 210 square feet of surface per horse-power for an engine working day and night. The depth in that case is quite unimportant; but if the engine is to work only say twelve hours per day, the surface area might be reduced, because the water would cool during the night, but in that case depth or capacity becomes a necessity, as we will proceed to show, although the question is a very complicated one.

(201.) Let us assume that 82° shall still be the *mean* temperature of the condensation water; this, however, from the nature of the case, will not now be uniform, but will be lowest in the morning after cooling down all night, rising all day till it reaches its maximum at the end of the day's work.

If we admit a variation of 10° each way, we shall have 72° and 92° for the minimum and maximum temperatures of the condensation water. The difference of the temperatures of the water entering the reservoir from the engine and departing to it will still be 40°; for if the engine receives the water 10° colder, it will also return it 10° colder than before, so that in the morning at starting, the water in the reservoir will be 72° at one end and 112° at the other, and at night 92° and 132°. The mean loss of heat per square foot by col. 12 of Table 79, between 72° and 112° is 222 units, and between 92° and 132° it is 421 units per hour: the mean loss during the day is therefore (222 + 421) ÷ 2 = 332 units per hour. During the night, when the engine is not working, the temperature will become uniform from end to end, being (92 + 132) ÷ 2 = 112° at first, and (72 + 112) ÷ 2 = 92° at last. Between these two temperatures, 92° and 112°, the mean loss of heat by col. 12 is 294 units per square foot per hour.

Now the total heat given out by the engine in the twelve working hours, or $65790 \times 12 = 789480$ units, has to be divided into two unequal parts, having the ratio of 332 to 294, hence we have $789480 \times 332 \div (332 + 294) = 418700$ units to be dissipated during the day, or $418700 \div 12 = 34900$ units per hour, and the mean rate of loss being as we have seen 332 units per square foot, we shall require $34900 \div 332 = 105$ square feet of surface per nominal horse-power.

The question now is, what must be the capacity of the reservoir, or the quantity of water necessary to hold the heat which accumulates during the day, namely, $789480 - 418700 = 370780$ units. The temperature of the water being raised 20°, we shall require $370780 \div 20 = 18539$ lbs., or $18539 \div 62 \cdot 3 = 298$ cubic feet of water, and as we have an area of 105 square feet, the depth would be $298 \div 105 = 2 \cdot 84$ feet if the reservoir has vertical sides: with sloping sides of course the depth would be greater, which is a matter of calculation.

(202.) We find as the general result of this investigation, that when an engine works day and night the depth is unimportant, and that we require 210 square feet of surface per nominal horse-power. When the engine works only twelve hours per day, 105 square feet will suffice, but in that case the depth must be such as to give 298 or say 300 cubic feet per horse-power.

Table 80 gives a general comparison of these rules with satisfactory cases in practice: it is remarkable that by an accidental coincidence, the conditions as to temperature, &c., that we assumed, give for twelve hours, precisely half the area per horse-power required for twenty-four hours per day. This fact is very convenient for those numerous cases where the work cannot be remitted for more than a very brief period, such as water-works for the supply of towns, and where for safety *two* engines are used instead of one of double power, so that in case of break-down or stoppage for repairs, one of the engines working day and night may do the work usually performed by two working twelve hours per day. Table 80 shows that the reservoir will do equally well in either case, as illustrated at Brighton and Sutton.

TABLE 80.—Of CONDENSATION RESERVOIRS; Cases in Practice.

	No. of Engines.	Total Nominal Horse-power.	Hours worked per Day.	Surface Area in Square Feet.			Capacity in Cubic Feet for 12 Hours per Day.	
				Actual.	Calculated.			
					24 Hours.	12 Hours.	Actual.	Calculated.
Brighton..	2	200	12	14464	..	21000	87884	60000
Ditto ..	1	100	24	14464	21000	..	..	..
Sutton ..	2	40	12	4500	..	4200	14742	12000
Ditto ..	1	20	24	4500	4200	..	..	..
Ramsgate	1	30	12	3240	..	3150	9228	9000
Sevenoaks	1	20	12	2240	..	2100	7152	6000

(203.) "*Dryness of Air Increased by Heat.*"—The capacity of air for carrying moisture increases very rapidly with the temperature, as shown by col. 7 of Table 68; thus, at 202° saturated air holds about one hundred times the weight of water that would saturate air at 32°. If air at 32° saturated and holding by col. 9 of the same table ·00379 lb. of vapour per pound of air, be suddenly heated to 42°, it will no longer be saturated, because at this latter temperature it could hold ·00561 lb.; it has therefore only ·00379 ÷ ·00561 = ·68, or 68 per cent. of the vapour that would saturate it, and would show by Table 70 about 37° on the wet-bulb thermometer (182) or 5° of cold. Table 67, which is of more extensive application, would have given the same result, thus the forces of vapour in saturated air at 32° and 42° are ·181 and ·267 inch respectively, and if saturated air at 32° be raised to 42° it will contain only ·181 ÷ ·267 = ·68, or 68 per cent. of the moisture that would saturate it.

The philosophy of drying and evaporation by heated air depends on this fact: in our dwellings too, the damp external air entering them is not only warmed, but is at the same time converted from damp to dry air, and our sense of comfort arises from both causes.

If air at 32° and saturated with moisture be successively raised from 32° to 72°, we should obtain the dryness, &c., given by Table 81. Conversely, the relative humidity of air

is increased by reducing the temperature; thus with half-saturated air at 62°, the force of vapour by Table 67 is ·556 ÷ 2 = ·278 inch, which is the force of vapour in fully saturated air at 43°, and if cooled below that point the air can no longer hold the whole of its moisture, but deposits it as dew upon adjacent objects; this is therefore called the dew-point.

TABLE 81.

Temp. of Air.	Temp. of Wet-bulb.	Cold.	Degree of Saturation.	State of the Air.
°	°	°		
32	32	0	1·00	Saturated.
42	37	5	·68	Moderately dry.
52	42	10	·46	Dry.
62	46	16	·32	Very dry.
72	49	23	·23	Intensely dry.

(204.) The dryness of air as the result of heating it, sometimes becomes objectionable; thus by Table 81 air at 32° heated to 62° would have its state of humidity 32 per cent. only, which would be intolerable. With our open fires we seldom get an objectionable degree of dryness, the *air itself* is seldom raised to a high temperature, because the only useful heat given out by an open fire is radiant heat which passes through the air without raising its temperature (311) (278), and is absorbed by the walls, which afterwards heat the air moderately. Besides, the quantity of air drawn in by the open throat of the chimney is about six times the amount required for the combustion, and the dryness is kept down by the large volume of damp external air passing through the room. But with close stoves the *air* becomes much more highly heated, as also where steam or hot-water pipes are used, and a very objectionable amount of dryness would ensue if means were not employed to prevent it. This is usually done by placing a vessel of water on the stove, where it becomes heated and gives out vapour copiously.

(205.) "*Evaporating Vessels for Stoves, &c.*"—Say we take the case of a stove in which 112 lbs. of coals are burnt in ten hours, and that we have a vessel one square foot in area

placed on it and maintained at 142°, while the air in the room is at 62°, and external air 32° and saturated. For 11·2 lbs. of coals per hour we require $11 \cdot 2 \times 300 = 3360$ cubic feet, or $3360 \times \cdot 0807 = 271$ lbs. of air, which, at 32° and saturated, will contain $\cdot 00379 \times 271 = 1 \cdot 03$ lb. of vapour. But during the hour, the evaporating vessel will add, by col. 3 of Table 79, ·634 lb. of vapour, and we shall then have $1 \cdot 03 + \cdot 634 = 1 \cdot 664$ lb. of vapour, whereas 271 lbs. of air at 62° would require, by col. 9 of Table 68, $\cdot 01179 \times 271 = 3 \cdot 195$ lbs. of vapour to saturate it; it therefore holds $1 \cdot 664 \div 3 \cdot 195 = \cdot 52$, or 52 per cent., whereas without such a vessel we should have had ·32 only, as per Table 81, which would have been uncomfortably dry. With a vessel of double area we should have had 1·268 lb. of vapour added, and the air would then have contained $1 \cdot 03 + 1 \cdot 268 = 2 \cdot 298$ lbs. of vapour, or $2 \cdot 298 \div 3 \cdot 195 = \cdot 72$, or 72 per cent., which would have been too damp, &c.

(206.) "*Evaporation at the Boiling Point.*"—When a liquid is not injured by a high temperature, evaporation by *boiling* is the most economical of all, as shown by Table 79: thus, by col. 10 the heat at 212° is only 1186 units per pound, instead of 2750 units, as at 62°. But this is with an open vessel exposed to radiation, &c., and when the evaporation is effected without positive ebullition: if ebullition be allowed a close cover may be used, leaving only a small aperture for the escape of steam, the losses of heat by the air and radiation in cols. 7 and 8 are suppressed, and we only require 966 units as by col. 9, which is only $966 \div 2750 = \cdot 35$, or 35 per cent. of the heat at 62°. In that case a large surface is not necessary; the evaporating pan may have any convenient form, such as Figs. 59, 60. Say we take Fig. 60 with a double or steam case, the inner vessel containing 100 gallons of water, and having the dimensions given by Table 83, the area in *contact with the water*, or the real heating surface, being about 17 square feet. With say 15 lbs. steam, having by Table 71 a temperature of 250°, we have $250 - 212 = 38°$ difference between the steam and the boiling water, and by Table 85, col. 2, with 1° difference, and water at 212°, the rate of transmission of heat is 1000 units per square foot of

heating surface per hour, and we have in our case $(17 \times 1000 \times 38) \div 966 = 670$ lbs., or 67 gallons of water at 212° evaporated to steam per hour.

(207.) We have here supposed that the water was at 212° to begin with; if the water had been cold the case would have been modified. Say that the water was at 110° and we require to know the quantity that will be raised from that temperature to 212° and evaporated in twenty minutes, or one-third of an hour. To heat 100 gallons, or 1000 lbs., $212 - 110 = 102°$, we require $1000 \times 102 = 102000$ units of heat; the mean temperature while being heated will be $(212 + 110) \div 2 = 161°$, and the mean difference of temperature between the steam and the water $= 250 - 161 = 89°$. The rate of evaporation varies very much with the varying temperature, as shown by Table 85, rising from 248 units at 110° to 1000 units at 212°; to find the mean rate we may take a mean of all the numbers in col. 3 between the two extreme temperatures, but col. 4 will give the same result more readily; thus we have $(5522 - 1456) \div 10 = 405$ units per square foot per hour, or $405 \times 17 \times 89 = 612765$ units per hour in our case, and the water will be raised from 110° to 212° in $102000 \div 612765 = \cdot167$ hour, leaving $\cdot333 - \cdot167 = \cdot166$ hour to do the evaporation work, and as by (206) we had 67 gallons per hour, we now have $67 \times \cdot166 = 11\cdot1$ gallons heated from 110° to 212°, and evaporated in 20 minutes.

CHAPTER VIII.

DISTILLATION.

(208.) "*Principles of Distillation.*"—It will be seen by Table 10, that the boiling points of liquids differ considerably, for instance, water boils at 212°, and alcohol at 173°; if a mixture of these two fluids is heated to 173°, the alcohol in it would rise in vapour, leaving the water behind, and thus in theory the whole of the alcohol might be separated, and the whole of the

water remain. But practically, it is found that part of the water becomes *entangled* with the alcoholic vapour, and passes away with it; by repeating the process, however, the alcohol may be obtained almost pure.

(209.) The vapour thus evaporated is conducted into a refrigerator, where it is condensed and restored to the liquid state again. The distilling apparatus consists therefore essentially of a closed evaporating vessel, and a condensing vessel, connected together by a pipe to conduct the vapour. The evaporator might be heated direct by the fire, or by steam-pipes circulating through the liquid to be evaporated, &c.

(210.) In this country alcohol is commonly distilled from wort made by the infusion of barley, it contains about 10 per cent. of alcohol; by the evaporation of this to one-half, nearly the whole of the spirit passes over, the liquor thus obtained being composed of one-fifth spirit and four-fifths water. Say we had a still containing 100 gallons of wort; we should obtain by the first distillation 50 gallons of liquor composed of 10 gallons of alcohol and 40 gallons of water. The work to be done may be divided into three portions—namely, to evaporate from say 60°, 10 gallons, or $10 \times 8 \cdot 13 = 81 \cdot 3$ lbs. of alcohol; to evaporate from 60°, 40 gallons, or $40 \times 10 = 400$ lbs. of water, which passes over with the spirit; and to raise the 50 gallons of water, which remains in the still at the end of the operation, from 60° to 212°. By the rules in (18) we have:—

		Units.
Evaporation of the Alcohol ..	$(908 - 60) \times \cdot 622 \times 81 \cdot 3 =$	42845
" " Water	$(1178 - 60) \times 400$ $=$	447200
Heating the Water which remains	$(212 - 60) \times 500$ $=$	76000
	Total	566045

The sizes of the still necessary to do this work may be found by analogy with a steam-boiler. By (118) 70,000 units are equal to 1 horse-power, and if we had to do the work in our case in an hour we have $566045 \div 70000 = 8$ horse-power, for which by Table 49 we require about 121 square feet of effective boiler surface, and 7·6, say 8 square feet of fire-grate. The

consumption of fuel would be about $566045 \div 6000 = 95$ lbs. of coal (112).

The fuel might be economized by heating the wort beforehand from 60°, the assumed temperature of the atmosphere, to, say 212°, the temperature of the still, by a heater in the flue or otherwise. The 100 gallons of wort contains 10 gallons of alcohol and 90 gallons of water; for the alcohol we have $(212 - 60) \times \cdot 622 \times 81 \cdot 3 = 7684$ units; and for the water $(212 - 60) \times 900 = 136800$ units; altogether 144,484 units, which is equal to $144484 \div 6000 = 24$ lbs. of coal saved by thus utilizing waste heat, the consumption being reduced to $95 - 24 = 71$ lbs.

(211.) "*Condensing Apparatus.*"—The apparatus described in (234) for heating liquids by steam, may be applied equally well for the condensation of vapour in the process of distillation, and the sizes necessary for any particular case may be found by the same rules.

The most common form of condenser is a helix or worm in a vessel of cold water, as in Fig. 61. The work to be done in our case (210) is to absorb the heat that had been employed in evaporating the 400 lbs. of water and 81·3 lbs. of alcohol, or $447200 + 42845 = 490045$ units. The temperature of the cold water may be say 60°, which entering at the bottom of the vessel containing the worm becomes heated by the coil to say 120° at the top, where it departs. The mean temperature is therefore 90°, at which, by Table 85, the loss for 1° difference of internal and external temperature is 220 units per square foot per hour. The mean difference in our case is $212 - 90 = 122°$, hence we have $122 \times 220 = 26840$ units per square foot, and we require $490045 \div 26840 = 18 \cdot 2$ square feet of surface. If the pipe is $1\frac{3}{4}$ inch diameter outside (319), its circumference is ·46 foot, and the length must be $18 \cdot 2 \div \cdot 46 = 40$ feet.

(212.) The quantity of cold water required for condensation will vary with the difference of temperature at entry and exit. If the temperature of the water in the vessel was uniform throughout, or 90° in our case, the rise of temperature would be $90 - 60 = 30°$, and we should require $490045 \div (30 \times 60) = 272$ lbs., or 27·2 gallons of cold water per minute. But the

water will not naturally have a uniform temperature, indeed it would require continual stirring to make it so; the water at the bottom where it should enter will be 60°; in the middle it will be 90°, and at the top where it departs it will be 120°; it has therefore been heated $120 - 60 = 60°$, and we should require in that case $490045 \div (60 \times 60) = 136$ lbs., or 13·6 gallons per minute.

CHAPTER IX.

ON DRYING.

(213.) "*Drying in Open Air*."—This is the commonest and cheapest mode of all, but in our climate it is too uncertain to be depended on for many manufacturing purposes; it is frequently used however in drying paper, glue, whiting, &c. In such cases a covered building is used to keep off the rain, the sides being open to admit the air freely, but shutters are provided to keep out the air as much as possible on damp days. The rate of drying varies exceedingly, see (188), (189); frequently the air is only about half saturated with humidity, and drying proceeds rapidly, especially with a wind; but in winter it is often completely saturated, and the drying process is completely suspended. The laws by which drying is governed are explained in the chapter on evaporation.

(214.) "*Drying by Heated Air*."—The capacity of air for moisture increases rapidly with the temperature, as we have shown in (203), and the efficacy of hot-air drying depends on that fact. The philosophy of the process will be best understood by an illustration.

Let Fig. 67 be a drying closet, in which the air entering at the bottom becomes highly heated by contact with the steam-pipes, and rising through the closet, finally escapes by the chimney at the top. Say we had 10 lbs. of water to evaporate from 42°; that the external air of a November day was at 42° and *completely saturated* with moisture, the exit

temperature being say 102°, and let us admit for illustration that the air at exit is only *half* saturated with moisture.

Now, every pound of air at entry and at 42°, is by col. 9 of Table 68 charged with ·00561 lb. of water, and at its exit at 102° with ·04547 lb. if saturated, but in our case $\cdot 04547 \div 2 = \cdot 02273$ lb. only; it therefore takes up in its passage $\cdot 02273 - \cdot 00561 = \cdot 01712$ lb. of water, and to carry off 10 lbs. we shall require $10 \div \cdot 01712 = 584$ lbs. of air.

(215.) To evaporate 10 lbs. of water from 42° requires by (18) $(1178 - 42) \times 10 \div 11360$ units of heat, or the amount that would raise 11,360 lbs. of water 1°, and the specific heat of air (5) being ·238, this is equal to $11360 \div \cdot 238 = 47731$ lbs. of air 1°. But we have only 584 lbs. of air to do the work required, it must therefore be heated $47731 \div 584 = 82°$, and the air must enter the closet at $102 + 82 = 184°$.

Thus we have 584 lbs. of air, heated from 42° to 184°, which coming in contact with the wet clothes is cooled down to 102°, the heat thus parted with serving to evaporate the 10 lbs. of water.

The total quantity of heat expended in the process is not only the 11,360 units required to evaporate the water, but also that required to raise 584 lbs. of air from 42° to 102°, which is equal to $584 \times (102 - 42) \times \cdot 238 = 8340$ units, making a total of $11360 + 8340 = 19700$ units.

(216.) "*Position of Outlet Opening, &c.*"—In Fig. 67 we have shown the common mode of arranging the inlet and outlet openings, but the plan is a very bad one, as may be seen in the chapter on ventilation (334, &c.); the heated air takes the shortest course to the chimney, and escapes only partially saturated with moisture, and in those parts of the drying room out of the course of the current the drying process proceeds very slowly. If instead of entering by numerous openings uniformly distributed as in the figure, the air entered at one place, the case would be still worse.

The best position for the outlet opening is near the floor, the requisite draught being obtained by a chimney, which may be made of light woodwork. The heated air rises in a body to the roof of the closet, as in (337), whence it is gradually drawn down by the action of the chimney; all the

horizontal layers of air will have the same temperature throughout, as in Fig. 93, and the drying in every part of the room will be equally effective.

(217.) "*Drying Closet for Linen, &c.*"—Figs. 68 to 70 give a good arrangement of a drying closet for linen; the horses are sometimes made of wrought iron, but well-seasoned deal is the lightest, cleanest, and best material in all respects; they are formed with a back and front plate, A B, about 12 inches wide, connected together by horizontal rails, on which the linen is suspended. They are supported by two flanged wheels, running on iron rails C, which are prolonged outside the closet, far enough to allow the horse to be drawn out, which is done by the handle D, the upper part being guided by a long wooden rail E E, which is also prolonged outside the closet. The openings, F, are about an inch narrower than A B, and say 2 inches less in height, so that when the horse is in its place, the opening is closed by the plate B, and when drawn out, by the plate A. The air entering from without by the channel G, fills the chamber H, and rises by a series of holes into the chamber J, containing 21 — 9-feet lengths of 4-inch steam-pipes. The floor of the closet is closed all over, except from L to M, which is covered by an open grating of cast iron, the full width of the room, and through this the heated air rises in a body to the roof, where it is distributed, and descends at the back part of the room to the opening N, by which it escapes into the chimney O, and thence into the atmosphere.

(218.) The length of steam-pipes including bends is about 210 feet of 4-inch pipe; if we take the pressure of steam at 10 lbs., its temperature by Table 71 will be about 240°, and at that temperature an *enclosed* pipe as in our case will yield by Table 90, 314 units per foot run, or $314 \times 210 = 65940$ units per hour; admitting that 10 per cent. is lost by radiation, &c., from the walls of the closet, we shall have $65940 \times \cdot 9 = 59346$ units available for drying purposes.

We will assume that the air leaves the steam-pipes and enters the closet at 172° and departs at 82°; it is first heated by the steam-pipes 130°, or from 42° to 172°, and the available

heat is sufficient to raise $\frac{59346}{130 \times \cdot 238} = 1914$ lbs. of air to the required temperature.

(219.) By Table 68, a pound of air at 42° saturated with moisture contains ·00561 lb. of water, and at 82, ·02361 lb.; it therefore takes up in passing through the closet ·02361 − ·00561 = ·018 lb. of air, and we have in our case ·018 × 1914 = 34·5 lbs. of water absorbed from the linen per hour. If the wet linen is introduced at 50°, each pound of water in it will require 1178 − 50 = 1128 units of heat to evaporate it, and in our case 1128 × 34·5 = 38923 units per hour. This heat has to be supplied by the air in the act of cooling from 172° to 82°, and we must see that it is capable of doing it; it will yield 1914 × (172 − 82) × ·238 = 41000, or rather more than is necessary.

If we had assumed 162° for the temperature at entry, we should then have had $\frac{59346}{120 \times \cdot 238} = 2078$ lbs. of air heated, which would carry off ·018 × 2078 = 37·4 lbs. of water, requiring 37·4 × 1128 = 42187 units of heat. The air in cooling from 162° to 82° would yield 2078 × (162 − 82) × ·238 = 39565 units only, being 42187 − 39565 = 2622 units too little. The conditions assumed in (218) are therefore nearly correct, and the power of the closet may be taken at 34·5 lbs. of water per hour.

(220.) "*Air-chimney, &c.*"—We have in our case 1914 lbs. of air per hour, or ·55 lb. per second, and by Table 24, this is equal at 42° to ·55 ÷ ·0791 = 6·95 cubic feet at entry from external air, and at 82° to ·55 ÷ ·0733 = 7·5 cubic feet at exit. But to this last has to be added the 34·5 lbs. of vapour taken up in the closet: this is only 34·5 ÷ 3600 = ·0096 lb. per second, or ·0096 × 21·07 = ·2 cubic foot (79), and the volume of air at exit is increased by it to 7·5 + ·2 = 7·7 cubic feet per second.

The mean temperature of the air in the closet is (172 + 82) ÷ 2 = 127°, the air in the chimney is at 82°, and both are saturated with moisture. It is necessary to remember this last fact, because the relative weights are affected by the presence of vapour, and the draught-power of the chimney is affected

thereby. We assumed in (214) for the sake of varying the illustration, that the air departed only *half* saturated, but it is a necessary condition where economy is considered that the air should be saturated.

(221.) The condition of the chimney and closet with reference to the creation of a draught current is peculiar, and must be understood before we can calculate the necessary sizes of openings, &c.

Let Fig. 71 be an outline diagram, in which A is the chimney and B the closet, both of the same height, the air being at 82° in the chimney, and 127° in the closet. The air in B, being lighter than in A in consequence of its higher temperature, would ascend, and motion would ensue in the direction shown by the arrows, being the reverse of what is required.

Let Fig. 72 be a similar diagram, with a chimney twice the height of the closet; the column in the chimney is now opposed not only by the column in the closet, but also by another one in the imaginary chamber C, which makes up the height of the chimney with air at the external temperature of 42°. The *mean* temperature of the combined column B C is therefore $(42 + 127) \div 2 = 84°\cdot5$, that in A remaining at 82°; with this height we should therefore still have a reverse draught.

In Fig. 73 we have a chimney three times the height of the closet, and the column in the chimney is now opposed by a column of equal height composed of B, C, D, whose mean temperature will be $(127 + 42 + 42) \div 3 = 70°$, or 12° colder than the air in the chimney, and we shall now obtain a draught in the right direction.

(222.) We can now calculate the power of the chimney in our case, and will assume for it a height of 28 feet, or four times the height of the closet. The fact that the air at 42°, 82°, and 127°, is all saturated with moisture complicates the case, and it will not be quite correct to take a mean temperature for the columns B, C, D, &c., as we have done with dry air in (221).

The weight of air saturated with vapour is given by col. 8 of Table 68; for 127° we must interpolate between 122° and 132°, and we have $(\cdot065042 + \cdot063039) \div 2 = \cdot06404$ lb. per

cubic foot, and for the closet 7 feet high $\cdot 06404 \times 7 = \cdot 44828$, the weight of the column in B; for C, D, &c., we have $\cdot 07884 \times 21 = 1\cdot 65564$, and together $\cdot 44828 + 1\cdot 65564 = 2\cdot 104$. The air in the chimney at 82° weighs (as a column 1 foot square) $\cdot 07226 \times 28 = 2\cdot 023$ lbs.

Then by (150), &c., the 28 feet of the lighter air is equal to $(28 \times 2\cdot 023) \div 2\cdot 104 = 26\cdot 92$ feet of the denser, leaving $28 - 26\cdot 92 = 1\cdot 08$ foot to produce velocity, which by the laws of falling bodies will give $\sqrt{1\cdot 08} \times 8 = 8\cdot 3$ feet per second theoretically.

Admitting that by the loss of effect from friction, change of direction, and successive enlargements and contractions of the air-passages through the closet (163) which it would be impossible to calculate, this velocity is reduced to half (394) (397), we shall have 4·15 feet per second in the chimney and openings generally; that through the closet would be less, because of the greatly increased area there.

(223.) The chimney O and the opening N having to pass, as we have seen in (220), 7·7 cubic feet of air per second, must have an area of $7\cdot 7 \div 4\cdot 15 = 1\cdot 85$ square foot, and may be $1\cdot 5 \times 1\cdot 25$ foot; the channel G and the holes in the top of the chamber H must be $6\cdot 95 \div 4\cdot 15 = 1\cdot 67$ square foot, &c., &c.

(224.) To facilitate the establishment of the draught when the closet is first heated, it will be well to have an opening P, by which the heated air can pass direct from the steam-pipes into the chimney; when the draught is well established this must be closed, otherwise we should have a waste of heat, and the drying operation would be retarded. A sliding door worked by a rod outside would be the most convenient mode of regulating the size of the opening.

(225.) It will be seen, that with damp air, the chimney must not be less than four times the height of the closet. Where such a height is impracticable from local reasons, a low one may be made to answer by keeping the opening P permanently open, the effect being to increase the temperature of the air in the chimney; but this is an expensive mode of effecting the purpose, and where possible, a high chimney should always be secured.

(226.) "*Drying Closets for Asylums, &c.*"—In Asylums, Schools, and similar establishments, the size of the drying closet must be fixed by experience; it will vary considerably with the character of the inmates, &c., Lunatic and Pauper Asylums requiring of course a larger washing and drying apparatus than others. Generally, we may allow 1 square foot of drying horse for children, say $1\frac{1}{2}$ for men, and 2 for women, estimating the area of the horse as its length multiplied by its height; thus in our case Fig. 68, &c., we have $8 \cdot 5 \times 6 \times 5 = 255$ square feet of surface, which would suffice for say 250 children; or $255 \div 1 \cdot 5 = 170$ men; or $255 \div 2 = 128$, say 130 women.

In the case of a school for 1200 pauper children of both sexes near London, the work had to be done with thirteen horses, each $9 \cdot 5 \times 6 \cdot 5$ feet, which by our rule would have sufficed for $9 \cdot 5 \times 6 \cdot 5 \times 13 = 802$ children only. It was therefore much too small, and to compensate for that fact, very long hours had to be worked; in damp weather when the whole of the work had to be done by the closet, from 6 A.M. to 9 P.M. for six days per week scarcely sufficed to accomplish it.

An ordinary blanket weighing $3\frac{1}{4}$ lbs. when dry contains about $6\frac{1}{2}$ lbs. of water when wrung as dry as possible; the closet Fig. 68 would contain five such blankets, and to evaporate the $32 \cdot 5$ lbs. of water in them, we should require $32 \cdot 5 \div 34 \cdot 5 = \cdot 94$ hour, say 1 hour. A common sheet weighing $1\frac{3}{4}$ lb. dry, contains 2 lbs. water, and five such, holding 10 lbs. of water, would be dried in $10 \div 34 \cdot 5 = \cdot 29$ hour, or rather more than a quarter of an hour, &c.

(227.) The weight of water remaining in woven fabrics varies very much with the different means adopted for its expulsion. Table 82 gives the results of M. Rouget's observations, from which it will be observed that hand-wringing, which is the method most destructive to the fabric, is also the least effective in the result, the weight of water that remains in calico and linen being three times the amount left by the hydro-extractor. But to obtain these results required from 500 to 600 revolutions per minute of a cylinder 31 inches diameter.

TABLE 82.—Of the WEIGHT of WATER in WOVEN FABRICS after WRINGING, &c., the Weight when Dry being 1·0.

	Flannel.	Calico.	Silk.	Linen.
After hand-wringing	2·0	1·0	·95	·75
After a powerful press ..	1·0	·6	·5	·4
After rotary hydro-extractor	·6	·35	·3	·25

(228.) "*Drying at High Temperatures.*"—It is stated in (189) that if access and free motion of the air employed in carrying off the vapour in evaporating and drying operations be prevented by even a loosely fitting cover, evaporation ceases and the drying process is stopped. Iron founders and others, however, use a closed room as a drying stove for their cores and moulds; a large coke fire being kindled in an open iron cage in the centre, the moulds are piled in, the loosely fitting iron doors are closed, and no more air is admitted than is just sufficient to keep up a slow fire, the products of combustion escaping by the cracks in the doors, &c. At first sight this seems to be wrong in principle, but the fact is there is very good and deep philosophy in it. If a common drying closet were used, with a free current of air at a comparatively low temperature as in (214), the *surface* of the mould would be quickly dried, but the interior would remain wet for an indefinite time, and when the molten metal was poured into it the intense heat would find out the internal moisture, convert it into steam, destroy the mould, and endanger the workmen. With a closed stove such as described above, the mould is either thoroughly dried through, or not dried at all: the drying process does not proceed sensibly until the mould is heated to 212° or the boiling point of water, and then the whole of the water in it is converted into steam, and the mould is at once dried to its centre. It will be seen from this that a current of air is essential only for temperatures below 212°: most drying operations are carried on below that temperature. Obviously there would be no advantage in temperatures much superior to 212°.

(229.) "*Drying in Closed Room.*"—Drying may be effected

at low temperatures without heat or ventilation in a closed room, or even in a vacuum, by placing with the articles to be dried some material that has a strong affinity for moisture, such as chloride of calcium or high-dried (228) oatmeal. The rapidity of absorption at first will depend simply on the surface area of the absorbent exposed to the air, but its continuance will depend on the thickness or mass: in either case the rate of absorption will decrease with time until the absorbent is fully saturated with moisture.

Oatmeal, dried at 350°, $\frac{5}{8}$ inch deep, exposed to air at a mean temperature of 50°, and humidity ·7, was found to absorb in each of four successive hours 125, 83, 63, and 42 grains per square foot per hour respectively; in 1, 2, 3, 4 hours, 125, 208, 271, and 313 grains were absorbed, so that to absorb 1 lb. in those times, we require 56, 34, 26, and 23 square feet $\frac{5}{8}$ inch deep. In each of seven successive days, 1130, 580, 250, 120, 90, 40, 40 grains per square foot per day were absorbed, so that in 1, 2, 3, 4, 5, 6, 7 days 1130, 1710, 1960, 2080, 2170, 2210, and 2250 grains per square foot were absorbed, equivalent to 4·5, 6·8, 7·8, 8·3, 8·6, 8·8 and 9 per cent. of the weight of dry meal, and to absorb 1 lb. in those times we require 6·2, 4·1, 3·6, 3·4, 3·226, 3·167 and 3·111 square feet respectively.

Chloride of calcium gave similar results, but was much more enduring in its action; in 1, 2, 3, 4 hours, 78, 156, 234, and 312 grains per square foot were absorbed, the thickness being $\frac{5}{8}$ inch, air 48°, humidity ·75. In each of seven successive days 1368, 1017, 958, 918, 900, 802, and 703 grains respectively were absorbed, so that in 1, 2, 3, 4, 5, 6, 7 days, 1368, 2385, 3343, 4261, 5161, 5963, and 6666 grains per square foot were absorbed, equivalent to 6·0, 10·5, 14·7, 18·8, 22·7, 26·3, and 29·4 per cent. of the weight of dried absorbent. To absorb 1 lb. in 1, 2, 3, 4 hours, we require 90, 45, 30, 22½ square feet, and to absorb 1 lb. in 1, 2, 3, 4, 5, 6, 7 days, 5·1, 2·9, 2·1, 1·64, 1·36, 1·17 and 1·05 square feet $\frac{5}{8}$ inch deep.

When a given quantity of water has to be absorbed in less than four hours, oatmeal is the most effective: if in four hours, both absorbents are alike in power, but for four days chloride

of calcium has double the power of oatmeal, and three times its power for seven days.

It should be observed, that in these experiments the humidity of the air was nearly constant; as applied to a drying room it would be variable: also that the power of the saturated absorbents is easily restored by re-drying at a high temperature (228).

(230.) *Drying Cylinders.*"—In paper-making, calico-printing, and other machines, the drying is conveniently effected by passing the fabric over polished cylinders of cast iron, &c., heated internally by steam. Clement found that a single thickness of calico in contact with a copper plate heated by steam at 212° was dried at the rate of 1·45 lb. of water per square foot of drying surface per hour M. Chameroy found the rate to be 1·8 lb. with one thickness, and ·91, or about half, with two thicknesses of cloth passing over a cylinder. By many experiments of M. Royer twenty pieces of calico as they came from a press weighed 330 lbs., and after drying, 167 lbs., having lost 163 lbs. of water, but the steam condensed in doing that work was 224 lbs., the practical result being $163 \div 224 = \cdot 73$, or 73 per cent. of the heat utilized, and therefore 27 per cent. wasted in radiation, &c., which is unavoidable, for the work occupied 3½ hours, and the drying cylinder exposes a large surface to cold air and radiation. Usually, however, this loss is unimportant, for the cost of this method of drying is *nothing;* where a high-pressure steam-engine is used, the exhaust steam instead of escaping into the atmosphere, is made to pass through the drying cylinders and give out its latent heat there. In such a case a common high-pressure engine is more economical than the best condensing engine, the steam being used twice over — driving the engine — and heating the drying cylinders.

In another case a machine with six cylinders in a badly closed room and external air near the freezing point, the weight of steam condensed was double the weight of water evaporated, hence 50 per cent. was wasted.

CHAPTER X.

ON HEATING LIQUIDS.

(231.) "*Heating Liquids by Fire direct.*"—Liquids may be heated directly by a fire, or by steam, which may be applied in several ways. When water is heated by a fire the best position for the fire is immediately beneath the vessel, and the worst possible position is the top, for when water is heated it expands, becomes lighter, and ascends, being replaced by colder water, which in its turn is heated, and so on until the whole mass is raised at once to the required temperature.

(232.) The form of the vessel is unimportant, only it must be such as to receive the heat of the fire readily; Figs. 59 and 60 are common and convenient forms, and for convenience of calculation the dimensions given are for a capacity of 1 gallon, and the dimensions for any other capacity may be found by multiplying by $\sqrt[3]{\text{gall. required}}$; thus if a copper to hold 125 gallons was required, the $\sqrt[3]{\ }$ of 125 is 5 and the sizes would be given by multiplying all the dimensions in the Fig. 59 or 60 by 5.

(233.) For small vessels of this form we cannot reckon on much more than the radiant heat in the coals, or 6500 units per pound by Table 44, and allowing that 10 per cent. is wasted, we have $6500 \times \cdot 9 = 5850$ units per pound of coal, or $5850 \div 966 = 6$ lbs. of water at 212° to steam.

TABLE 83.—Of the PROPORTIONS of BOILING and EVAPORATING PANS.

Capacity of Copper in Gallons.	Dimensions of Shallow Copper as per Fig. 59.					Dimensions of Deep Copper as per Fig. 60.						Area of Fire-grate in Square Feet.	Coals burnt per Hour in lbs.	Pounds of Water at 212° to Steam per Hour.	Gallons of Water heated from 62° to 212° per Hour.
	A	B	C	D	E	F	G	H	J	M	N				
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.				
20	5½	8¼	1½	15¼	24½	4	14½	1½	20	22	20¼	½	5	29	15·5
50	7½	11½	1½	20½	34	6	20	1½	27½	30	26½	1	10	56	30·0
100	9½	14	1½	25	42	7	25	1½	33½	37	33	1½	15	87	46·5
150	11	16	2	29	48	8	28	2	38	42	38	2	21	124	66·2
200	12	18	2	32	53	9	31	2	42	47	42	2¼	24	144	77·0
250	13	19	2	34	57	9½	33	2	44½	51	45	2½	29	173	92·3
300	14	21	2	37	60	10	36	2	48	54	48	3	31	185	98·8

We may allow 18 square feet of fire surface per cubic foot of water evaporated: thus with Fig. 60, for 300 gallons capacity, we have a gross area of about $12 \cdot 5 + (13 \cdot 3 \times 3) = 52 \cdot 5$ square feet, which will evaporate $52 \cdot 5 \div 18 = 2 \cdot 92$ cubic feet, or $2 \cdot 92 \times 62 \cdot 32 = 185$ lbs. of water at 212°, requiring $185 \div 6 = 31$ lbs. of coal, and $31 \div 10 = 3$ square feet of fire-grate, &c. Table 83 gives the sizes of coppers up to 300 gallons calculated in this way.

(234.) "*Heating Liquids by Steam.*"—There are three methods commonly used for heating liquids by steam: by forming a steam-jacket, or double vessel, as in Fig. 60; by a worm circulating through the liquid and filled with steam, as in Fig. 61; and by blowing the steam by a pipe direct into the liquid to be heated.

(235.) Péclet gives an experiment with an apparatus of the form like Fig. 60, in which 1980 lbs. of liquid (beetroot juice) at 39° was raised to 212° in 16 minutes, the steam being at 30 lbs. per square inch (above the atmosphere), and consequently at 274°, and the surface exposed to its action, $25 \cdot 8$ square feet. The work done per hour is $\dfrac{1980 \times (212° - 39°) \times 60}{16 \times 25 \cdot 8} =$ 50000 units nearly per square foot. The *mean* temperature of the water in this case was $(39° + 212°) \div 2 = 125°$, and the *difference* of temperature between that and the steam, $274° - 125° = 149°$; and we have therefore from this experiment $50000 \div 149 = 335$ units per square foot per hour for a difference of 1°.

(236.) An experiment was also made with a worm apparatus like Fig. 61, the pipe was 138 feet long and $1 \cdot 34$ inch diameter outside, having a surface of 48 square feet, and was filled with steam at 274°. The vessel contained 880 lbs. of water at 46°, which was heated to 212° in 4 minutes, and in 11 minutes more 550 lbs. of water were evaporated.

(237.) In the first case the work done per hour in heating the water was $\dfrac{880 \times (212° - 46°) \times 60)}{4 \times 48} = 45650$ units per square foot for a mean difference of $274° - \left(\dfrac{212 + 46°}{2}\right) = 145°$, and for 1° we have $45650 \div 145 = 315$ units.

(238.) In the second part of the operation the work done was $\frac{550 \times 966 \times 60}{11 \times 48} = 60400$ units per hour per square foot; the difference between the temperatures was constant, and equal to $274° - 212° = 62°$, therefore for 1° difference we have $60400 \div 62 = 974$ units.

(239.) In another experiment with a similar apparatus two worms were used, each 49 feet long, 1·34 inch diameter outside, presenting altogether a total exterior surface of 39·5 square feet, filled with 15 lbs. steam having a temperature of 250°, and 132 lbs. of water were evaporated in five minutes, The difference of temperature being $250° - 212° = 38°$, we have $\frac{132 \times 966 \times 60}{5 \times 39 \cdot 5 \times 38°} = 1020$ units for 1° difference per square foot per hour.

(240.) An experiment was made by myself, for Easton and Amos, of London, with a thin welded tube of wrought iron $1\frac{1}{4}$ inch diameter outside and about $\frac{1}{16}$ inch thick, fixed vertically in a vessel of water 12 inches square and 3 feet 7 inches deep in water as in Fig. 62, steam was turned gently on at A, and the cock B was kept a little open to carry off any water that primed from the boiler, &c.; an air-vent was left open constantly at E, and the water condensed was discharged by an open nozzle at C, and collected in a vessel, D. We can estimate the amount of heat transmitted in two different ways, namely, by the rise in the temperature of the water in F, and by the weight of water condensed, and there should be perfect agreement between the two. The weight of the water in F was 223 lbs., and it was stirred well to produce uniformity of temperature; the surface of the tube in contact with the water was 1·4 square foot. In one experiment the water was raised from 65° to 110°, or 45° in 15 minutes, and 10·218 lbs. of water were collected at D, the quantity at D, calculated from the rise in temperature of the water, should have been $(223 \times 45) \div 966 = 10 \cdot 38$ lbs., agreeing very nearly with the experimental quantity. The *mean* temperature of the water was $(65 + 110°) \div 2 = 87° \cdot 5$; and the difference between that and the steam

$212° - 87°\cdot5 = 124°\cdot5$; and we have therefore $\frac{223 \times 45 \times 60}{15 \times 124\cdot5 \times 1\cdot4}$ = 230 units per square foot per hour for a difference of 1°. Two other experiments gave 207 and 210 units respectively, and calculating from the water condensed, rather less in all three cases. Table 84 gives the collected results.

TABLE 84.—Of the HEATING POWERS of a VERTICAL TUBE with STEAM.

Weight of Water in the Vessel.	Temperature of the Water.				Water collected at D in 15 minutes.		Error.	Units per Square Foot per Hour for 1° difference of Internal and External Temperature.
	Min.	Max.	Mean.	Increase.	Actual.	Calculated.		
lbs.	°	°	°	°	lbs.	lbs.	lbs.	
223	65	110	87½	45	10·218	10·38	·162	230
223	60	102½	81¼	42·5	8·67	9·81	1·14	207
223	69	109½	89¼	40·5	8·95	9·35	·4	210

(241.) These experimental results appear to be very anomalous, varying as much as 1020 ÷ 207 = 5 to 1 nearly, and even with the same apparatus (237) (238) and in the course of the same experiment as 974 ÷ 315 = 3·1 to 1. This seems, however, to be due simply to the well-known fact, that as water is heated it becomes more lively in its movements, the internal currents by which the heat is carried off increasing in rapidity as the temperature rises, the result being that the particles of water in contact with the heating surface are more rapidly renewed and more heat is given out. By plotting the experiments in a diagram we obtain a curve from which we have derived Table 85, applying which to the experiments, the anomalies disappear. It is necessary to distinguish between the effect with a fixed temperature of water and that with a variable one; for the latter, the mean of the effects at each temperature between the two extreme points must be taken from col. 3. Thus with the vertical tube between 65° and 110°, we have 196, 204, 214, 226, and 240 units, the mean being 216 units: experiment gave 230 units. This result might have been obtained more easily from col. 4; thus from 40° to 110° being 1456, and from 40° to 60° = 376 units, there-

fore from 60° to 110° the mean comes out (1456 − 376) ÷ 5 = 216 units as before. For the experiment in (235), where the temperature varied from 39° to 212°, we must take a mean of the whole of col. 3 in Table 85, and we obtain 325 units per square foot for 1°, as in col. 5. Table 86 gives a general view of all the experiments, col. 8 having been calculated by Table 85. There appears to be practically no difference in effect, between a steam-pipe and a steam-cased vessel of the ordinary form. (206).

TABLE 85.—Of the POWER of STEAM-JACKETED VESSELS, and STEAM-PIPES in HEATING WATER.

Temperature of Water, &c., to be Heated.	Units per Square Foot per Hour for 1° difference, Steam and Water.		Between 40° and the different Temperatures.	
	At each Temperature.	Between each Temperature.	Sum.	Mean.
°				
40	185		000	000
		186		
50	188		186	186
		190		
60	193		376	188
		196		
70	200		572	191
		204		
80	209		776	194
		214		
90	220		990	198
		226		
100	233		1216	203
		240		
110	248		1456	208
		256		
120	265		1712	214
		274		
130	284		1986	221
		294		
140	305		2280	228
		316		
150	328		2596	236
		340		
160	353		2936	245
		371		
170	389		3307	254
		406		
180	430		3713	265
		462		
190	494		4175	278
		547		
200	600		4722	295
		800		
212	1000		5522	325
(1)	(2)	(3)	(4)	(5)

(242.) The third method of heating fluids with steam by blowing it direct into the water to be heated, is commonly used for rough purposes, there are some objections to it for refined use; where the steam-boiler is connected with a steam-engine particles of grease may prime over with the steam; the volume of liquid is augmented by the steam condensed to a considerable extent, thus to heat 100 gallons of water from 32° to 212° (or

TABLE 86.—Of EXPERIMENTS on the POWER of STEAM-CASED VESSELS and STEAM-PIPES, in HEATING WATER.

Temperature of the Water heated.			Temp. of the Steam.	Difference of Temperature of Steam and Water.	Units per Square Foot per Hour for 1° difference of Temperature.				Kind of Heater.
					By Experiment.		By Table.		
Minimum.	Maximum.	Mean.			Units.	Mean.	Units.	Mean.	
°	°	°	°	° °					
65	110	..	212	147 to 102	230		216		Vertical Tube.
60	102½	..	212	152 „ 109½	207	216	210	216	Ditto.
69	109½	..	212	143 „ 102½	210		221		Ditto.
89	212	..	274	235 „ 62	335	325	325	329	Steam-cased Vessel.
46	212	..	274	228 „ 62	315		333		Worm } same apparatus.
..	..	212	274	62°	974	997	1000	1000	Ditto } same apparatus.
..	..	212	250	38	1020		1000		Ditto.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	

180°) will require $100 \times 10 \times 180 = 180000$ units of heat, to obtain which we must condense $180000 \div 966 = 186$ lbs. of steam, or nearly 19 gallons, which will of course be added to the water to be heated. Another objection is the loud noise with which this method is accompanied; this may be partially obviated by placing the end of the steam-pipe in a small vessel full of small fragments of broken granite, &c., a cover of coarse wire-gauze being stretched over the mouth of the vessel to prevent the stone being scattered by the steam.

When these objections are not serious this method should be used in preference to any other; it is not only the most simple but also the most economical mode of heating by steam, the whole of the heat contained in the steam being given out to the water.

(243.) "*Heating Liquids by Gas, Petroleum, &c.*"—By (387) the total heat in coal-gas is 696 units per cubic foot, nearly the whole of which should be utilized by a properly constructed apparatus in which the gas-jets are almost completely surrounded by the water, and the outlet for the products of combustion is a pipe of considerable length traversing the water to be heated. Allowing that 10 per cent. is wasted, we have $696 \times \cdot 9 = 626$ units utilized, and with gas at 4*s.* per 1000 this is equal to $626 \times 1000 \div 48 = 13042$ units for a penny: comparing this with the cost of coal, and allowing, as in (389), 44,800 units for a penny, we have the ratio $44800 \div 13042 = 3 \cdot 4$ to 1. But in practice the great cost of gas is reduced by the fact that the consumption commences and terminates with the operation; where a coal fire has to be maintained all day to perform occasional work which could be done with gas in three or four hours, gas may be cheaper than coals.

With the common open gas-stoves in which the vessel is placed over jets or wire-gauze, there are losses by radiation, &c., which reduce the useful effect very considerably: with a small portable stove, having a wire-gauze 3 inches diameter, and a vessel 8 inches diameter, the mean useful effect in raising water from a low temperature to the boiling point was found by experiment to be 340 units per cubic foot, or $340 \times 1000 \div 48 = 7143$ units for a penny, or $44800 \div 7143 = 6 \cdot 27$ times the cost

of coals. The consumption of gas with this stove was about 10 cubic feet per hour; hence $340 \times 10 \div 160° = 21$ lbs., or say 2 gallons of water might be heated from 52° to 212° per hour. The area of 3 inches being 7 square inches, we have $340 \times 10 \div 7 = 486$ units per square inch of wire-gauze per hour.

The total heat in petroleum is by (59) 20,240 units, and if we admit, as with gas, that 90 per cent. is utilized with the most perfect apparatus, we have $20240 \times \cdot 9 = 18200$ units per pound: a gallon weighing 8·4 lbs. and costing say eighteenpence, this is equal to $18200 \times 8\cdot 4 \div 18 = 8493$ units for a penny, or $44800 \div 8493 = 5\cdot 27$ times the cost of coals.

(244.) "*Heating Liquids at the Top.*"—It has been generally admitted that water and other liquids will not conduct heat downwards, because when heated, liquids become lighter and remain persistently in contact with the heating surface, whereas when heated at the bottom the heat is rapidly carried upward through the mass of the liquid by the lighter stratum rising to the surface.

To test the accuracy of this statement, a flat-bottomed cylindrical vessel of tin plate 7 inches diameter outside, $3\frac{1}{2}$ inches deep, was filled with hot water to a depth of $3\frac{1}{4}$ inches, and placed with its bottom just touching the surface of a volume of cold water, so large that its temperature was nearly constant, being raised only 3°, or from 52° to 55°, by the heat received from the experimental vessel. The time occupied in cooling down successive 10° was observed as given in col. 5 of Table 87. Part, however, of the heat thus lost would be due to radiation from the surface of the water and the sides of the vessel, to ascertain the amount of which, the experiment was repeated with the bottom of the vessel in contact with a thick pad of blotting-paper, which being by Table 101 one of the worst conductors of heat, the loss by the bottom would practically be suppressed, and the time to cool down 10° was found to be as in col. 4. The difference of time between cols. 4 and 5 is due to the heat conducted downwards by the water, but some analysis is necessary to obtain the results we require. Thus from 170° to 160°, or 10°, required 269″ in air, and 110″ on water, and we have to determine how much of the 10° was due to the air, &c.

and how much to the water. In air the loss was 10° in 269″, therefore in 110″ the loss from air alone would have been $10 \times 110 \div 269 = 4°\cdot09$, hence $10 - 4\cdot09 = 5°\cdot91$ was due to the water alone. The weight of water cooled was 4·27 lbs., and the area of the bottom being ·2672 square foot, the loss of heat is $4\cdot27 \times 5\cdot91 \div \cdot2672 = 94\cdot4$ units per square foot in 110″, or $94\cdot4 \times 3600 \div 110 = 3090$ units per hour, and the difference of temperature between the water in the vessel, and that in contact with its bottom being $165 - 53 = 113°$, we have finally $3090 \div 113 = 27\cdot35$ units per square foot per hour for 1° difference. We thus obtain the experimental results in col. 8, and by plotting them in a diagram we eliminate the irregularities of experiment, and we thus find the mean results in col. 9, which shows that the loss of heat is not constant, but increases with the temperature from 9·3 units per degree at 85°, to 34·2 units at 195°, or about 3·7 to 1. Comparing this with Table 85, it appears that the heat transmitted downwards by a flat surface is at 85° only about $\frac{1}{23}$rd part of the heat given out by an ordinary steam-coil, or steam-jacketed vessel: at 195° the ratio is $\frac{1}{18}$th.

TABLE 87.—Of EXPERIMENTS on the Power of STILL WATER in conducting Heat downwards.

Temp. of the Hot Water.	Mean.	Diff. of Temp. in Vessels and in Bath.	Time in Seconds to cool down 10°.		Due to the		Units per Sq. Foot per Hour for 1° diff.	
			In Air.	On Water.	Air.	Water.	By Expt.	By Diagram.
° 200	°	°			°	°		
190	195	143	141	60	4·26	5·74	38·60	34·2
180	185	133	176	78	4·43	5·57	30·93	32·0
170	175	123	227	92	4·05	5·95	30·30	29·7
160	165	113	269	110	4·09	5·91	27·35	27·3
150	155	102	355	142	4·00	6·00	23·84	25·1
140	145	92	424	178	4·20	5·80	20·32	22·8
130	135	81	581	218	3·75	6·25	20·32	20·6
120	125	71	793	292	3·68	6·32	17·40	18·3
110	115	61	993	370	3·73	6·27	16·00	16·0
100	105	50	1355	518	3·83	6·17	13·71	13·8
90	95	40	1980	777	3·92	6·08	11·29	11·5
80	85	30	2870	1200	4·18	5·82	9·33	9·3
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)

CHAPTER XI.

ON HEATING AIR.

(245.) "*Heating Air, &c.*"—Air cannot be heated direct by radiant heat for reasons given in (278), but radiant heat can be absorbed by solid bodies, which afterwards give out that heat to the air in contact with them. In our dwellings heated by open fires, radiant heat alone is given out usefully by the fire, the rest being wasted by passing off up the chimney; this radiant heat is absorbed by the walls, and while part of that passes through the walls and is dissipated on external objects, the other part serves to warm the air of the room (311). In heating air therefore the object is to place heated solid bodies which will impart their heat to the surrounding air.

(246.) "*Heating Air by Stoves.*"—Stoves supply a cheap and economical method of obtaining heat; nearly the whole of the heat which any fuel is capable of yielding may be utilized by using a long flue-pipe conveying the products of combustion to the outer air. As much as 95 per cent. of the gross total heat in coals may be thus obtained (250), whereas with steam-boilers we seldom realize more than 63 per cent., as shown by Table 47.

There are, however, several serious objections to stoves, especially for small rooms; a long flue-pipe is unsightly, and on that account inadmissible in many cases; when made of cast iron with the fuel in contact with it on one side and the air of the room on the other, the air is burnt and vitiated to an intolerable degree; and in any case the air is apt to become uncomfortably *dry* for reasons given in (204). In the north of Europe, where stoves are used exclusively for all purposes, they are made of brickwork with thick walls, therefore of large dimensions.

(247.) "*Horizontal Flue-pipes.*"—The heat given out by a stove-pipe varies with the temperature from end to end, being of course greatest at the end next to the stove where the heat is greatest. Let Fig. 11 be a stove-pipe of great length, for the

purpose of illustration, heated to a dull red heat of 1230° at one end and 150° at the other, or exit end. Dividing the length into six portions with equal differences of temperature there will be (1230 − 150) ÷ 6 = 180° difference between point and point, and the series of temperatures will be 1230°, 1050°, 870°, 690°, 510°, 330°, and 150°, the mean of the intervals being 1140°, 960°, 780°, 600°, 420°, and 240°, and the excess over the external air at 60° will be 1080°, 900°, 720°, 540°, 360°, and 180°, as in Fig. 11. Then by the rules in (282), (313), &c., we have for a 4-inch horizontal pipe, the results given by Table 88.

TABLE 88.—Of the HEAT given out by a HIGHLY HEATED HORIZONTAL STOVE-PIPE of IRON, 4 inches diameter, exposed to Air at 60°.

Mean Temp. of the Pipe.	By Radiation to the Walls.					By contact to the Air.					Total per Square Foot per Hour.
	R.	Diff.	Ratio.	Units.	Per Cent.	A.	Diff.	Ratio.	Units.	Per Cent.	Units.
°		°					°				
1140	(·7 ×	1080 ×	23·00) =	17388	or 92 and	(·5745 ×	1080 ×	2·450) =	1520	or 8 =	18908
960	(·7 ×	900 ×	12·68) =	7988	„ 86 „	(·5745 ×	900 ×	2·348) =	1214	„ 14 =	9202
780	(·7 ×	720 ×	7·17) =	3614	„ 79 „	(·5745 ×	720 ×	2·230) =	922	„ 21 =	4536
600	(·7 ×	540 ×	4·19) =	1584	„ 71 „	(·5745 ×	540 ×	2·085) =	647	„ 29 =	2231
420	(·7 ×	360 ×	2·56) =	568	„ 59 „	(·5745 ×	360 ×	1·897) =	392	„ 41 =	960
240	(·7 ×	180 ×	1·61) =	203	„ 55 „	(·5745 ×	180 ×	1·615) =	167	„ 45 =	370
				Mean = 74					Mean = 26		

The heat lost between point and point in Fig. 11 is equal, namely, 180°, but the loss *per square foot* is very unequal, as shown by the table; thus at the end next the stove 18,908 units per square foot are lost per hour, and the distance from A to B necessary to reduce the temperature of the pipe 180° would be very short, but at the exit end, only 370 units per square foot are lost, and the distance F, G must be greater to effect the same reduction of 180°, in fact it must be 18908 ÷ 370 = 51·1 times the length A, B. Then if the first length A, B is 1 foot, the next B, C must be 18908 ÷ 9202 = 2·05 feet; the next C, D = 18908 ÷ 4536 = 4·17 feet, &c., &c., as in Fig. 11. The total loss of heat will now be the same for each length; for instance, the first length loses 18,908 units and the last 370 × 51·1 = 18908 units also, and so on throughout. The loss by the whole pipe will be 18908 × 6 = 113448 units, or

113448 ÷ 86·12 = 1317 units per foot run: we have here assumed that the surface area of a 4-inch pipe is 1 square foot per foot run, which is near enough to the truth for practical purposes.

(248.) It should be observed, that not only does the total amount of heat given out vary greatly with the temperature from end to end, but also that the *proportions* into which it divides itself between the *walls* and the *air* vary greatly with the temperature: at the stove end 92 per cent. of the total heat emitted by the pipe is given out by radiation to the walls, and only 8 per cent. to the air; but at the exit end the heat is nearly equally divided, the walls receiving 55 and the air 45 per cent. Taking the whole length of the pipe, the walls receive 74 per cent. and the air 26 per cent. of the heat emitted. At still lower temperatures the air would receive half the heat or even more (316). When therefore the object is to heat the walls rather than the air, which is sometimes the case (311), (373), the temperature of the pipes should be high, and stove-pipes are more effective than hot-water, or low-pressure steam pipes. The effect of the latter may be seen by Table 88, the lowest temperature in col. 1, or 240°, is that due to 10¼ lbs. steam by Table 71, and the heat is divided into 55 per cent. to the walls and 45 per cent. to the air. The heat thus received by the walls is usually divided again into two parts, one part heating the air in contact with the wall, and the other passing through the wall to the outer surface, where it is finally dissipated and wasted.

(249.) To apply these results to practice: by (60) the gross total heat in a pound of coals is 13,000 units, which by (98) is reduced to 12,350 units nett, of which say one-fourth, or 3090 units, is given out by the body of the stove, leaving 9260 units to be given out by the flue-pipe. The air being heated 1170°, or from 60° to 1230°, the weight capable of carrying that heat is 9260 ÷ (1170 × ·238) = 33·3 lbs., and as this passes out of the pipe at 150°, or 90° above the atmosphere, it still retains 33·3 × 90 × ·238 = 713 units of heat which is wasted, thus leaving 9260 − 713 = 8547 units per pound of coal to be given out by the surface of the pipe. We found in (247) with the

conditions there assumed, the mean rate of heat given out was 1317 units per square foot, we shall therefore require $8547 \div 1317 = 6 \cdot 5$ square feet of surface per pound of coal. Then, with a stove burning say 10 lbs. of coal per hour we require $6 \cdot 5 \times 10 = 65$ square feet, and with say a 7-inch pipe the length would be $65 \div 1 \cdot 84 = 35$ feet.

Table 88 is calculated for a 4-inch pipe, but may be applied within moderate limits to pipes of other sizes without serious error.

(250.) With these proportions only $713 \div 13000 = \cdot 055$, or $5\frac{1}{2}$ per cent. of the gross total heat in the fuel, is wasted and $94\frac{1}{2}$ per cent. is utilized, but to obtain so good a result, a large surface of flue-pipe was necessary. If the air be permitted to escape at a higher temperature than that we have assumed, namely, 150°, the heat given out per square foot would be greatly increased, and the surface area of the pipe might be reduced. This is shown by Table 89; thus with 330° instead

TABLE 89.—Of the EFFECT of STOVE-PIPES at different TEMPERATURES.

Temp. at Stove End of Pipe.	Temperature at Exit End of Pipe.						Lbs. of Air per lb. of Coal.
	1050°	870°	690°	510°	330°	150°	
	Total Units of Heat per Square Foot per Hour.						
1230°	18908	12400	7856	4934	2700	1317	33·3
1050	..	9202	6080	3958	2223	1111	39·3
870	..	..	4536	3079	1774	910	48·1
690	..	..	..	2231	1360	719	61·8
510	..	..	..	..	960	534	86·5
330	..	..	..	..	..	370	144·0

of 150° at exit, the mean heat per square foot is more than doubled, becoming 2700 instead of 1317 units per hour, and the surface might be reduced to half. But in that case, the air would escape at $330 - 60 = 270°$ above the atmosphere, carrying off $33 \cdot 3 \times 270 \times \cdot 238 = 2138$ units, or $2138 \div 13000 = \cdot 16$ or 16 per cent. of the gross total heat in the fuel, and only 84 per cent. would in that case be utilized. If the air enters

from the stove at 1230°, and be allowed to escape successively at 150°, 330°, 510°, 690°, 870°, and 1050°, as in Table 89, the heat wasted would be respectively $5\frac{1}{2}$, 16, 27, 35, 49, and 60 per cent. of the gross total heat in the coals.

(251.) "*Vertical Flue-pipes to Stoves.*"—By (248) it is shown that at high temperatures the heat given out is principally that due to radiation, which is independent of form and position of the radiant (277), there will therefore be practically little difference between horizontal and vertical pipes.

Péclet made some experiments on vertical flues of cast iron, sheet iron, and earthenware, and obtained very anomalous results, cast iron appearing to emit three times as much heat as sheet iron ones under similar circumstances.

"*Vertical Cast-iron Flues.*"—A chimney of cast iron 54 feet high, 8 inches internal diameter, $\frac{4}{10}$ inch thick, total *outside* surface 122 square feet, was mounted on a small furnace, the temperature was taken at the top and bottom, and the velocity of the current was measured directly by the time which a whiff of smoke occupied in passing from the bottom to the top. The temperatures were 347° at the bottom, 170° at the top, external air 68°, and the velocity 14·8 feet per second.

The mean temperature of the air in the chimney was $(347 + 170) \div 2 = 258°$, its weight by Table 24, ·055 lb. per cubic foot, and the cross-sectional area being $8^2 \times \cdot7854 \div 144 = \cdot35$ square foot, we have $\cdot35 \times 54 \times \cdot055 = 1\cdot04$ lb. of air. This weight of air parts with $347 - 170 = 177°$ of heat, and its specific heat (5) being ·238, this is equal to $1\cdot04 \times 177 \times \cdot238 = 43\cdot8$ units of heat: this loss occurs in the time which a particle of air occupies in passing through the pipe, or in $54 \div 14\cdot8 = 3\cdot65$ seconds: in an hour therefore the loss would be $43\cdot8 \times 3600 \div 3\cdot65 = 43200$ units, or $43200 \div 122 = 354$ units per square foot per hour.

(252.) To calculate the heat given out by the rules (313) we may take R for new cast iron at ·648 from Table 95, and A at ·45 from Table 100. The mean temperature of the pipe being 258°, and the excess above external air $258 - 68 = 190°$, the ratio for radiation by Table 104 is 1·72, and for contact of air, 1·63 by Table 105, and the calculated loss comes out

$(\cdot 648 \times 190 \times 1\cdot 72) + (\cdot 45 \times 190 \times 1\cdot 63) = 351$ units per square foot per hour, or very nearly the experimental amount of 354 units.

(253.) "*Vertical Sheet-iron Flues.*"—Similar experiments were made with a sheet-iron chimney 52 feet high, $3\frac{1}{2}$ inches diameter, surface 48·6 square feet. The mean of eleven experiments gave: temperatures, at the bottom 536°, at the top 170°, external air 68°, velocity 9·8 feet per second.

The mean temperature of the air in the chimney is 353°, its weight ·049 lb. per cubic foot, and the cross-sectional area of the pipe being $3\cdot 5^2 \times \cdot 7854 \div 144 = \cdot 067$ square foot, the weight of the air in the chimney is $\cdot 067 \times 52 \times \cdot 049 = \cdot 17$ lb., which parts with $\cdot 17 \times (536 - 170) \times \cdot 238 = 14\cdot 8$ units of heat in $52 \div 9\cdot 8 = 5\cdot 3$ seconds, or $14\cdot 8 \times 3600 \div 5\cdot 3 = 10054$ units per hour, or $10054 \div 48\cdot 6 = 206$ units per square foot per hour.

Calculating as before (252) the value of R for ordinary sheet iron is ·5662; for $A = \cdot 48$; the ratio for radiation, 2·2; for contact of air 1·8, and the calculated loss becomes $(\cdot 5662 \times 285 \times 2\cdot 2) + (\cdot 48 \times 285 \times 1\cdot 8) = 601$ units per square foot per hour, or nearly three times 206 units, the experimental result. It is difficult to account for this great discrepancy, and the experiments must be regarded as of doubtful accuracy.

(254.) "*Height of Chimney for Stoves.*"—Considering the case in which the flue-pipe itself forms the chimney, we may have 1st, a uniform slope from end to end; 2nd, the part next the stove may be vertical and thereby act as a chimney while the rest is horizontal; and 3rd, the vertical part may be at the exit end. Taking the case in (249) we have a pipe 7 inches diameter, 35 feet long, which has to discharge $33\cdot 3 \times 10 = 333$ lbs. of air per hour, or 5·55 lbs. per minute; and it is the same throughout the length, but the volume varies with the varying temperature from end to end. At the stove end the *mean* temperature being 1230° by Fig. 11, we have by Table 24, $5\cdot 55 \div \cdot 0235 = 236$ cubic feet, and with 150° at the exit end $5\cdot 55 \div \cdot 0649 = 86$ cubic feet per minute. The discharge may be calculated by the rules in (155), &c., and it will be nearly correct if we take the arithmetical mean, or $(236 + 86) \div 2 =$

161 cubic feet for the friction, and the maximum volume or 236 cubic feet for calculating the head due to velocity at entry.

The head for friction with 161 cubic feet in a 7-inch pipe 11·7 yards long, by Table 61, is $\cdot 0000309 \times (161 \div 100)^2 \times 11\cdot 7 = \cdot 00094$ lb. per square inch, or $00094 \times 144 = \cdot 135$ lb. per square foot for friction alone. The area of a 7-inch pipe being ·267 square foot, the maximum velocity at entry will be $236 \div (\cdot 267 \times 60) = 14\cdot 7$ feet per second, the head for which by the laws of falling bodies with ·93 coefficient for contraction (153) is $\left\{14\cdot 7 \div (8 \times \cdot 93\right\}^2 = 3\cdot 9$ feet, which in air at the external temperature, say 62°, is by Table 24 equivalent to $3\cdot 9 \times \cdot 0761 = \cdot 297$ lb. per square foot for velocity alone, which added to that due to friction gives a total of $\cdot 135 + \cdot 297 = \cdot 432$ lb. per square foot.

(255.) If the flue were fixed with a uniform slope, the whole of it would be available as a chimney; in that case we must find the mean temperature of the internal air, which will not be the arithmetical mean of the extremes, or 690°, because although the temperatures decrease in that ratio, the lengths at each temperature are very variable, as shown by Fig. 11. If we multiply the mean temperature of each division A B, B C, &c., by its length, and divide the sum by the total length, we obtain 387° as an approximation to the true mean temperature of the air in the whole length. By Table 24, the weight of air at 387° is ·047 lb. per cubic foot, while the weight of the external air at 62° is ·076 lb., we have therefore for each foot in height of chimney an unbalanced pressure of $\cdot 076 - \cdot 047 = \cdot 029$ lb. per square foot, and the exit end must be $\cdot 432 \div \cdot 029 = 15$ feet higher than the stove end with uniform slope.

If the vertical part serving as a chimney is at the stove end, the mean temperature of that part would be about 1140°, as in Fig. 11, giving $\cdot 076 - \cdot 025 = \cdot 051$ lb. unbalanced pressure per foot in height, and we require $\cdot 432 \div \cdot 051 = 8\cdot 47$ feet to be vertical, and the rest, or 26·53 feet, might be horizontal.

If the vertical part is at the exit end, the mean temperature of that part would probably be about 240°, giving $\cdot 076 - \cdot 056$

= ·02 lb. unbalanced pressure per foot in height, and the vertical part must be ·432 ÷ ·02 = 21 feet high.

(256.) "*Heating Air by Steam-pipes, &c.*"—We shall see by (316) that the amount of heat given out by heated pipes to the surrounding air per square foot of surface, is not constant, but varies with the diameter, small pipes being most effective, and that 2, 3, 4 and 6 inch pipes yield respectively 327, 303, 291, and 279 units per square foot per hour, when the temperature of the pipe is 210°, and of the external air, &c., 60°. With thin metal such as is commonly used in practice, we may admit that the outside surface has sensibly the same temperature as the steam within, see Table 108 and (319), and assuming $\frac{1}{4}$, $\frac{5}{16}$, $\frac{5}{16}$, and $\frac{3}{8}$ inch for the respective thicknesses, we obtain ·654, ·95, 1·21, and 1·767 as the area or surface of the different pipes per foot run. Taking the 4-inch as an example, we find the loss of heat per foot run to be 291 × 1·21 = 352 units per hour; the specific heat of air (5) being ·238, this is equal to heating 352 ÷ ·238 = 1479 lbs. of air 1°, and as by Table 24 a cubic foot of air at 62° weighs ·0761 lb., we have 1479 ÷ ·0761 = 19435 cubic feet of air at 62°, heated 1° per foot run of 4-inch pipe per hour.

Table 90 has been calculated in this way, the loss per square foot for the different temperatures having been calculated as in (316).

"*Pipes enclosed in Narrow Chambers or Channels.*"—When pipes are enclosed in small chambers, whose sides being very near them, become highly heated, they emit very much less heat than when freely exposed to air and distant walls, having a low temperature. This case is investigated in (317) and (409), from which it appears that the loss in the case of an enclosed pipe is about 70 per cent. of the loss when freely exposed; and thus we obtain the numbers in the second part of Table 90.

(257.) "*Effect of Variations in Internal and External Temperature.*"—The Table 90 is calculated for pipes freely exposed in a room with air at 60°, but sometimes for drying and other purposes the air is at a much higher temperature, and at other times the temperature is much below 60°. The effect of these

TABLE 90.—Of the HEAT emitted by STEAM and HOT-WATER PIPES at different Temperatures, and differently exposed to Cooling Influences.

Mean Temp. of the Pipe.	Temp. of the Air.	Difference.	Units of Heat per Square Foot of Surface per Hour.				Units of Heat per Foot run of Pipe per Hour.				Pounds of Air heated 1° per Foot run of Pipe per Hour.				Cubic Feet of Air at 60° heated 1° per Foot run of Pipe per Hour.			
			in. 2	in. 3	in. 4	in. 6	in. 2	in. 3	in. 4	in. 6	in. 2	in. 3	in. 4	in. 6	in. 2	in. 3	in. 4	in. 6
PIPE FREELY EXPOSED TO AIR AND WALLS, BOTH AT ONE TEMPERATURE OF 60°.																		
300°	60°	240°	616	574	553	531	403	545	669	938	1693	2290	2811	3941	22235	28713	36919	51760
280	60	220	543	505	486	467	355	480	587	825	1491	2017	2466	3466	19582	26490	32387	45521
260	60	200	477	443	426	409	312	421	515	723	1311	1769	2164	3042	17218	23233	28421	39952
240	60	180	414	385	370	355	271	366	448	627	1138	1538	1882	2634	14946	20199	24717	34594
220	60	160	356	330	317	314	233	313	384	537	979	1315	1618	2256	12858	17271	21184	29629
210	60	150	327	303	291	279	214	287	352	493	900	1206	1479	2088	11826	15847	19435	27437
200	60	140	299	277	266	256	195	263	322	452	820	1105	1353	1900	10775	14507	17780	24967
190	60	130	271	251	241	231	177	239	292	408	744	1004	1227	1714	9777	13193	16123	22523
180	60	120	245	227	218	209	160	216	264	369	672	907	1109	1550	8830	11920	14573	20368
170	60	110	220	204	196	188	144	194	237	332	605	815	995	1395	7950	10710	13075	18330
160	60	100	196	181	174	167	128	172	210	295	538	722	882	1240	7070	9487	11590	16300
150	60	90	172	159	153	147	112	151	185	260	470	634	777	1092	6176	8331	10210	14350
PIPE ENCLOSED IN A NARROW CHANNEL, WITH BRICK SIDES, &c.: AIR AT 60°; WALLS 158°.																		
300	60	240	431	402	387	372	282	382	468	657	1185	1603	1968	2759	15564	20134	25843	36232
280	60	220	380	354	340	327	248	336	411	578	1043	1412	1726	2426	13707	18543	22671	31865
260	60	200	334	310	298	286	218	295	361	506	918	1238	1515	2129	12053	16263	19895	27966
240	60	180	290	270	259	248	190	256	314	439	794	1077	1317	1844	10462	14139	17302	24216
220	60	160	249	231	222	213	163	219	269	376	685	921	1129	1579	9006	12089	14829	20740
210	60	150	229	212	204	197	150	201	246	348	630	844	1032	1462	8278	11093	13604	19206
200	60	140	209	194	186	179	137	184	225	316	574	774	947	1330	7542	10155	12446	17477
190	60	130	190	176	169	162	124	167	204	286	521	703	859	1200	6844	9235	11286	15766
180	60	120	172	159	153	146	112	151	185	258	470	635	776	1085	6181	8344	10201	14258
170	60	110	154	143	137	132	101	136	166	232	424	571	697	977	5565	7497	9153	12831
160	60	100	137	127	122	117	90	120	147	207	377	505	617	868	4949	6641	8118	11410
150	60	90	120	111	107	103	78	106	130	182	329	444	544	764	4323	5832	7147	10010

changes on the amount of heat emitted is given by Table 91 for a great range of external and internal temperatures; thus say that we required 1000 cubic feet of air per minute, or 60,000 cubic feet per hour at 82°, the external air being at 32°; then by the table, we should require 60000 ÷ 211 = 284 feet of

TABLE 91.—Of the CUBIC FEET of AIR (at the Temperature of the Room) HEATED from the EXTERNAL TEMPERATURES, per hour per foot run of Steam-pipes, heated to 210°, and exposed to AIR, &c., having the Temperature of the Heated Room.

Temperature of the External Air.	Temperature of Air in the Room.						
	42°	52°	62°	72°	82°	92°	102°
	Cubic Feet of Air by 2-inch Pipe.						
12°	451	302	225	186	151	124	102
22	626	403	281	223	176	141	115
32	1253	604	375	279	211	165	131
42	..	1208	513	372	264	198	153
52	..	..	1126	557	352	247	184
	Cubic Feet of Air by 3-inch Pipe.						
12	560	406	313	250	204	168	138
22	840	502	391	300	238	192	156
32	1680	813	521	375	285	224	178
42	..	1626	782	500	354	268	208
52	..	..	1563	750	475	335	250
	Cubic Feet of Air by 4-inch Pipe.						
12	685	496	382	307	249	206	170
22	1028	661	478	369	291	235	191
32	2056	992	637	461	349	274	219
42	..	1984	955	615	437	329	255
52	..	..	1910	922	583	411	306
	Cubic Feet of Air by 6-inch Pipe.						
12	960	702	536	428	351	264	240
22	1440	936	671	514	409	330	270
32	2881	1404	894	643	491	385	309
42	..	2809	1342	857	614	462	360
52	..	..	2683	1285	818	578	432

2-inch pipe, or $60000 \div 285 = 210$ feet of 3-inch, or $60000 \div 349 = 172$ feet of 4-inch, or $60000 \div 491 = 122$ feet of 6-inch pipe, &c., &c.

(258.) "*Air-cocks, &c.*"—Water at ordinary temperature always contains a considerable quantity of air, which is expelled by heat, and passes off mixed with the steam, retarding the communication of the heat to the pipe containing it. It is therefore necessary to get rid of it, and this can be done most simply by small cocks fixed here and there along the course of the pipes, usually on the top; but air being heavier than steam at the same temperature and pressure, as shown by (177), theory indicates that the air-cock should be at the bottom of the pipe, and in one or two cases in which it has been tried, practice does not disprove it, so that the apparatus used to discharge the water condensed may sometimes be used to get rid of the air also.

(259.) "*Apparatus for discharging Condensed Water.*"—The quantity of water condensed by steam-pipes is shown by (316); it is very considerable, and being constant, the apparatus should be self-acting. Where the steam is taken from a low-pressure boiler, an inverted syphon, Fig. 50½, is the most simple contrivance for this purpose; but when the heating pipes are at the ground level, the syphon requires an excavation to be made for it, which may be difficult and sometimes impracticable; for 7 lbs. steam the column of water required is, by Table 38, about 16 feet, and allowing a margin for fluctuation in pressure, the syphon should not be less than 20 feet long for that pressure. There is a considerable loss of effect by the use of this apparatus, arising from the circumstance that the water is discharged at a very high temperature; in fact, at or near that of the steam, so that with steam at 210° and exterior air at 60° we might have $1178 - 60 = 1118$ units of heat per pound of steam; but if the water is discharged at 210°, we obtain only $1178 - 210 = 968$ units, or $968 \div 1118 = \cdot 86$, or 86 per cent. of the available heat, and 14 per cent. is wasted.

(260.) Figs. 51 and 52 show a simple apparatus by which this loss may be avoided, the water being retained till nearly cold; and it has the further advantage of discharging the air as well as the water: A is a ¾-inch gas-pipe about 8 or 9 feet long;

B B two rods of wrought iron about $\frac{5}{16}$ inch diameter, secured by nuts at the upper end to the winged socket D, which is fixed to the pipe by the pinching screw E, and similarly secured at the lower end to the cross-bar C, which carries a screw G by which the valve H is adjusted; those rods are guided by passing loosely through wings F F on the valve seat. The best position for this apparatus is the vertical one, as shown by Fig. 52; but where a vertical position is inconvenient, it may be fixed at only a slight angle with the horizon. The pipe A communicating with the bottom of steam-pipe J, will, when filled with steam, have the same temperature as that steam, and the screw G must then be regulated so as just to close the valve. The water that is condensed accumulates in A, and becoming cold, the pipe A is shortened, while the rods B B are not affected; the valve seat therefore retreats from the valve, which opens and allows the cold water to escape; but when hot water or steam follows, the pipe A, becoming heated, is expanded again to its normal length, and the valve is again closed, and so on. The variation in length is not very great, but with the length given is found by experience to be sufficient for the purpose. Taking the expansion of wrought iron from Table 16, and allowing that the water escapes at 110° while the steam is 210°, the expansion for a length of 108 inches is $\cdot 000006689 \times 108 \times 100 = \cdot 072$, or about $\frac{1}{14}$th of an inch. A brass tube about 6 feet long would expand about the same as an iron one 9 feet long, and may be used where a great length is objectionable.

(262.) Fig. 54 shows a valve box and float which is sometimes used for getting rid of the condensed water. A is a float of copper, &c., attached by a loosely-fitted ring or shackle B to the valve C; a ring of metal D is cast on the bottom, and serves to receive and retain the float in position; a similar ring E on the cover serves to prevent the float rising too high, and thereby drawing the valve out of its seat, &c. There is some difficulty in practice in keeping the float tight, especially with high-pressure steam, and the valve is apt to stick and give trouble. The steam, &c., enters at F, and the condensed water escapes at G.

(263.) It is very desirable that the water should not be

wasted, but wherever possible it should be conducted back to the boiler, and used over and over again; ordinary water contains salts of lime, &c., &c., which in the case of a steam-boiler do not pass off with the steam, but are deposited on the internal surface of the boiler, and are very destructive, by obstructing the transmission of the heat and causing the metal to become unduly heated; but by feeding the boiler with the water returned from the steam-pipes, which is of course pure distilled water, this may be avoided. In most cases where there is no steam-engine, a small donkey engine and pump is used; but latterly the Giffard's Injector has been used, and is by far the neatest and cheapest apparatus for this purpose.

(264.) "*Expansion of the Pipes.*"—Arrangements should be made to allow for the expansion caused by the great changes of temperature which steam-pipes experience; the simplest course is to mount them on rollers like Fig. 53, allowing them liberty at the ends to expand freely. In long ranges the expansion is considerable; thus with high-pressure steam at 300° and air at 50°, we have 250° difference between the extremes, and a pipe 100 feet long expands $\cdot 000006167 \times 250 \times 100 \times 12 = 1\cdot 85$ inch; with low-pressure steam at 210° and air at 50°, the expansion would have been 1·184 inch.

(265.) Where the movement of the pipe is inconvenient, an expansion joint must be used. Fig. 56 shows a good form for this; all the fitting parts should be roughly turned and bored, but should be slack fits; $\frac{1}{16}$th play may be allowed with advantage all over. C is a wrought-iron safety ring riveted on the end of the pipe A, the object of which is to prevent accident by the pressure of the steam blowing the joint apart; this pressure is considerable, for instance, with a 4-inch pipe, $\frac{1}{2}$ inch thick, and 45 lbs. steam, there is a force of 882 lbs. tending to blow the spigot out of its socket; and if by an oversight in fixing, a sufficient resistance is not provided, a serious accident may ensue. This cannot happen with Fig. 56, the wrought-iron ring C being prevented from coming out by a second ring D, which again is kept in position by the packing and gland bolts. Care should be taken in *fixing*, so as to adjust the length of the pipes, that C has room to move in the *right direction*; the effect of expansion

being to increase the length, C should be nearly touching D when cold, and the distance from C to E should of course be sufficient to allow for the utmost expected variation in length; if by carelessness C is fixed in contact with E, there can be no expansion, and the apparatus is useless.

(266.) "*Heating Air by Hot-water Pipes.*"—In Fig. 47 let A B C D be a pipe filled with water, and let heat be applied at A; the column A B will be heated and expanded, and becoming thereby lighter than the counterbalancing column C D, will begin to ascend in the direction shown by the arrow. Let us suppose that at A the water is heated to 210°, and that the air is at 60°, the pipe will give out heat to the air, so that the heat imparted to it at A is gradually and progressively lost; and assuming that it returns to A at 110°, and that it loses equal amounts of heat for equal distances from A, we obtain the series of temperatures given in the figure. We can calculate the velocity of the water in any particular case; say that the height A B was 40 feet, B C and A D 10 feet each, and the pipe 4 inches diameter, then by Table 21 we find the relative specific gravities of water at 190° and 140° to be about as ·9659 to ·9822, and the column C D being 40 feet, will be opposed by the column A B, having a weight equivalent to $40 \times \cdot 9659 \div \cdot 9822 =$ 39·336 feet; the two columns, therefore, do not balance one another, but there is a difference of $40 - 39 \cdot 336 = \cdot 664$ foot, or 8 inches, tending to produce motion in the water, and which by the principles of hydraulics would produce a velocity equal to the discharge of 70 gallons per minute. But 70 gallons, or 700 lbs. of water per minute cooled 100°, would yield $700 \times 100 \times 60 = 4200000$ units per hour, whereas by Table 90 a 4-inch pipe at a mean temperature of 160°, gives 210 units per foot, or in our case $210 \times 100 = 21000$ units per hour only, so that we have an excess of power, and to obtain the conditions shown by Fig. 47, which we assumed for the purpose of illustration, the pipe would have to be throttled by a valve or cock, &c.; or if left free, the water would return to the boiler at A at a temperature very little less than it left it, in which case the pipe being at a higher temperature throughout, would impart more heat than we found for it.

(267.) Fig. 48 shows a pipe precisely similar to Fig. 47, but here the fire is at B, half-way up the pipe; assuming the same temperatures as before, it will be seen by the figures that the *mean* temperature of both columns is exactly the same, or 160°; of course in such a case there is equilibrium, and no motion will ensue, nor will any forcing of the fire produce it. This will serve to show that in all cases the fire should be as near to the bottom of the ascending column as possible; when quite at the bottom, a very short column indeed will suffice to produce the necessary motion.

(268.) *Velocity necessary to renew the Heat."*—Let Fig. 49 be a pipe 500 feet long, 4 inches diameter; A B and C D each 1 foot, and let A B have a temperature of 210° and C D 160°, the water returning to B at 110°. At the given temperatures of 210° and 160°, Table 21 gives about ·9585 and ·9761 for the specific gravities of water, so that A B being 12 inches C D would be equal to 12 × ·9585 ÷ ·9761 = 11·783 inches; there would therefore be an unbalanced pressure of 12 − 11·783 = ·217 inch tending to produce motion. By Table 92, which

TABLE 92.—Of the FRICTION of WATER in PIPES.

Velocity in Feet per Second.	$\frac{H}{L} \times d$.	Theoretical Head to produce the Velocity in Column 1.	Velocity in Feet per Second.	$\frac{H}{L} \times d$.	Theoretical Head to produce the Velocity in Column 1.
		inches.			inches.
·01	·000008866	·0000187	·13	·0001943	·003168
·02	·00001870	·0000750	·14	·0002169	·003675
·03	·00002813	·0001687	·15	·0002394	·004118
·04	·00004148	·0003000	·2	·0003702	·007500
·05	·00005437	·0004687	·25	·0005266	·011717
·06	·00006830	·0006750	·3	·0007080	·016875
·07	·00008320	·0009187	·35	·0009154	·022968
·08	·00009920	·001200	·4	·001148	·030000
·09	·0001161	·001518	·45	·001406	·037970
·10	·0001341	·001875	·5	·001700	·046850
·11	·0001532	·002268	·55	·00200	·056720
·12	·0001732	·002700	·6	·00233	·06750

is calculated by Prony's formula, we may find the velocity which that small head would generate, for $\frac{H}{L} \times d$ is in our case

$\frac{\cdot 217}{6000} \times 4 = \cdot 000144$, which by the table is equal to a velocity of about ·105 foot per second, an extremely small velocity, but more than sufficient for the purpose, as we shall see presently. By Table 90 we find that a 4-inch pipe at 160° gives 210 units per foot run per hour; we have therefore in our case 210 × 500 = 105000 units per hour, or 105000 ÷ 3600 = 29·1 units per second, and each foot of water in a 4-inch pipe weighing 5·4 lbs., losing in our case 100°, this is equal to 5·4 × 100 = 540 units per foot, and to obtain 29·1 units, the water must move with a velocity of 29·1 ÷ 540 = ·053 foot per second; we found the velocity really obtained to be ·105 foot per second, or double the minimum necessary, so that with the given conditions, vertical columns only 1 foot high will work a pipe 500 feet long effectively.

(269.) To facilitate these calculations, we have given in Table 93 the head to be obtained with different temperatures of vertical columns of water in the descending pipe, that of the

TABLE 93.—Of the HEIGHT of EQUIVALENT COLUMNS of WATER at different TEMPERATURES, the Height at 212° being 12 inches.

Temperature of Water.	Height of Column in Inches.	Difference from 212° in Inches.	Temperature of Water.	Height of Column in Inches.	Difference from 212° in Inches.
° 212	12·000	·000	° 122	11·647	·353
202	11·954	·046	112	11·611	·389
192	11·908	·092	102	11·599	·401
182	11·868	·132	92	11·570	·430
172	11·824	·174	82	11·550	·450
162	11·783	·217	72	11·532	·468
152	11·746	·254	62	11·518	·482
142	11·710	·290	52	11·508	·492
132	11·677	·323	42	11·502	·498

ascending column being 212°. In Table 94 we have given the minimum velocity of current necessary in different sized pipes 100 feet long, calculated as in (268). It will be found that when the temperatures are constant, the velocity *necessary* varies in the simple ratio of the length of pipe, so that the table

applies to all cases. Thus a pipe 1000 feet long will require ten times the velocity for a pipe 100 feet long, &c.

TABLE 94.—For HOT-WATER PIPES, showing the Minimum Velocity of Current necessary for the Renewal of the Heat in a Pipe 100 feet long, with different Temperatures of Pipe, exposed to Air at 60°.

Temperature of the Water in the Pipe.				Diameter of Pipe in Inches inside.			
Mean.	As it leaves the Boiler.	As it returns to the Boiler.	Difference.	2	3	4	6
				Velocity in Feet per Second.			
°	°	°	°				
205	210	200	10	·3900	·2290	·164	·1040
200	210	190	20	·1993	·1190	·0817	·0510
190	210	170	40	·0904	·0535	·0367	·0230
180	210	150	60	·0545	·0326	·0224	·0139
170	210	130	80	·0368	·0219	·0151	·0094
160	210	110	100	·0262	·0156	·0107	·0067
150	210	90	120	·0191	·0119	·0786	·0049

NOTE.—The velocity necessary is simply proportional to the length of the pipe; thus a pipe 500 feet long requires five times the velocity given by the table for 100 feet, &c.

(270.) Table 92 will enable us to calculate the velocity of the current, having the head given by Table 93, &c., &c.

1st. Having the length, diameter, and velocity given, to find the *head*, take from Table 92 the number in column 2 opposite to the given velocity; this multiplied by the length of the pipe in inches, and divided by the diameter of the pipe in inches, will give the required head in inches, the height of column to give which must be calculated by Table 93.

2nd. To find the *velocity*, multiply the given head in inches (obtained by Table 93) by the diameter in inches, and divide by the length of the pipe in inches, and find the nearest number thereto in column 2, then opposite to that, in column 1, is the required velocity.

These rules do not include the head due to *velocity* (155) which is given by column 3 of Table 92. But this is so small that in most cases it may be neglected, except with *very* short pipes; thus the head due to ·053 foot velocity in (268) is only ·0005, or $\frac{1}{2000}$th inch of water.

(271.) We found in (268) that columns 1 foot high were

sufficient to work a pipe 500 feet long; but to obtain that result we were obliged to allow the water to return to the boiler very much reduced in temperature. The *mean* temperature of the pipe being low, we obtained from it only 210 units per foot, whereas we should have had 352 units if the water had retained throughout the temperature of 210°. It is desirable, therefore, to give long vertical columns where possible, as the temperature may then be higher, and the system more efficient. The velocity necessary for the renewal of the heat being proportional to the length, and the head or height of vertical columns proportional to the velocity squared, multiplied by the length, it follows that the height of column should be proportional to the cube of the length of the pipes, so that for lengths, 1, 2, 3, the column should be in the ratio 1, 8, 27.

(272.) Let Fig. 50 be a 3-inch pipe 400 feet long, and say that we require to find the height of the columns A and B for different temperatures; say A is 210° and B 190°, the mean temperature is then 200°. Table 94 gives ·119 as the necessary velocity for a pipe 100 feet long. In our case we require $\cdot 119 \times 4 = \cdot 476$ foot per second, which is between ·45 and ·5 in Table 92; we may take $\frac{H}{L} \times d$ in column 2 therefore at ·00155, which (270) multiplied by the length and divided by the diameter, both in inches, or in our case $\cdot 00155 \times 4800 \div 3 = 2 \cdot 48$ inches head. By Table 93, for 190° we have a head of ·092 inch for a column 1 foot high, therefore we shall require $2 \cdot 48 \div \cdot 092 = 27$ feet columns in our case; and the pipe will yield by Table 90, 263 units per foot run. Again, say A is 210° and B 170°, the mean temperature being 190°. Table 94 gives the velocity at $\cdot 0535 \times 4 = \cdot 214$ foot per second, which may be taken at ·0004 in column 2 of Table 92; and the head is $\cdot 0004 \times 4800 \div 3 = \cdot 64$ inch, which again by Table 93 is equal to $\cdot 64 \div \cdot 174 = 3 \cdot 7$ feet columns; the pipe will yield in this case for a mean temperature of 190°, 239 units per foot instead of 263 as we found it before.

(273.) "*Position of the Fire.*"—We have seen that it is desirable to place the fire as near the bottom of the ascending column as possible, and in (267) we have given a case that is impracticable where the fire was midway in the height of that

column. In some cases the excavation necessary for placing the fire at the bottom is very objectionable; and as it is *possible* to work a system of pipes with the fire considerably *above* them, by using lofty vertical columns, we will investigate the case, without, however, recommending its adoption where it can possibly be avoided.

(274.) Let Fig. 55 be a pipe 4 inches diameter, and altogether 200 feet long, of which the two columns A and B are each 40 feet high, and the pipes D, E, F, and G lying at the same level in the room to be heated, and let the fire be at C, 5 feet above the general level of the pipes in the room. If we allow that the water leaves the fire at 210° and returns at 190°, losing 20° in 200 feet, it follows that 1° is lost by each 10 feet in length; at H therefore it will be 189°·5; at J 210 − (35 ÷ 10) = 206°·5; and at K 206·5 − (40 ÷ 10) = 202°·5. The mean temperature from H to C will therefore be (189·5 + 190) ÷ 2 = 189°·75, from C to J = (210 + 206·5) ÷ 2 = 208°·25, and of the whole column A,

$$\frac{(189\cdot75 \times 5) + (208\cdot25 \times 35)}{40} = 205^\circ\cdot94, \text{ say } 206^\circ.$$

The mean temperature of the column B is (206·5 + 202·5) ÷ 2 = 204°·5 so that it is 1°·5 colder than A; and this is all the motive-power at our disposal; hence the critical character of such an arrangement.

By Table 21 the expansion of water between 202° and 212° is ·000411 for 1°, and a column of water 40 feet or 480 inches long, heated 1°·5, will expand ·000411 × 1·5 × 480 = ·296 inch, which is the head to produce motion. By Table 92, $\frac{H}{L} \times d$ is in our case $\frac{\cdot296}{200 \times 12} \times 4 = \cdot000493$, which by col. 2 may be estimated at ·24 foot velocity per second. The velocity necessary for the renewal of the heat in the water for a 4-inch pipe, whose mean temperature is 200°, we shall find by Table 94 to be ·0817 foot for a pipe 100 feet long, in our case therefore we shall require ·0817 × 2 = ·1634 foot per second, so that the velocity of ·24 foot actually obtained is sufficient.

CHAPTER XII.

ON THE TRANSMISSION OF HEAT, AND LAWS OF COOLING.

(275.) The cooling of heated bodies may be effected by three methods:—1st, by radiation; 2nd, by contact of cold air; and 3rd, by conduction. Putting U for the total loss from all causes, and R′, A′, and C′ for the respective losses by radiation, contact of air, and conduction, we have $U = R' + A' + C'$.

"*Loss of Heat by Radiation.*"—Let P, Fig. 74, be a plate say of building stone, 1 foot square and 1 inch thick, having both its surfaces, S, S′, as also the air in contact with S, at a constant temperature of 60°; and let the surface S be exposed to distant

TABLE 95.—Of the RADIATING and ABSORBING POWER of BODIES, being the Units of Heat emitted or absorbed per square foot per hour for a Difference in Temperature of 1° Fahr. From the Experiments of PÉCLET.

	Value of R.
Silver,* polished	·02657
Copper "	·03270
Tin "	·04395
Zinc and Brass, polished	·04906
Tinned Iron "	·08585
Iron, Sheet "	·09200
Lead "	·13286
Iron " ordinary	·5662
Glass	·5948
Cast Iron, new	·6480
Chalk	·6786
Cast and Sheet Iron, rusted	·6868
Wood Saw-dust, fine	·7215
Building Stone, Plaster, Wood, Brick	·7358
Sand, fine	·7400
Calico	·7461
Woollen Stuffs, any Colour	·7522
Silk Stuffs, Oil Paint	·7583
Paper, any Colour	·7706
Lamp-black	·8196
Water	1·0853
Oil	1·4800

* Copper silvered.

walls, W W W, maintained at 59°, or 1° lower than S. Under these conditions, the surface S will obviously lose no heat by contact of air, because that air has the same temperature as itself; for a similar reason, there will be no loss by conduction, because S′ has the same temperature as S; but S will send out in all directions towards W W rays of radiant heat, which will proceed in straight lines through the air, until they are intercepted and absorbed by the walls. The amount of heat thus lost will vary exceedingly with the nature of the surface. This is shown by Table 95, which gives the loss in units of heat per square foot per hour for 1° difference between S and W, as in our case; and for building stone this loss may be taken at ·736 unit per hour, &c. If W had the same temperature as S, radiation would cease altogether; if, on the other hand, the temperatures were reversed, S being in that case 1° *lower* than W W, it would absorb the same amount of heat as it emitted in the former case; the radiant and absorbing power of bodies being equal to one another.

(276.) For ordinary atmospheric temperatures of absorbing surfaces, say 50° to 60°, and small differences between S and W, say 30°, we may admit that the loss of heat is simply proportional to that difference; and that $R' = R \times (t - T)$, in which R' is the loss by radiation, in units per square foot per hour, R = the radiant power of the body from Table 95, t = the temperature of the radiant, and T = the temperature of the absorbent; but for high temperatures of T, and great differences between t and T, the loss of heat is much greater, following a complicated law, for which Dulong has given a rule that agrees well with experiment, up to a very high temperature; see (313).

(277.) The loss of heat by radiation is not affected by the *form* of the radiant body: a cube, a sphere, a cylinder, &c., will radiate the same amount of heat with equal areas under the same conditions, so long as the body is not of such a form as to *radiate to and from itself.* The colour of the surface seems to have no effect on the radiant power of bodies; at least this is true of paper and woven fabrics.

The radiation of heat is not affected by the distance of the absorbent: thus in Fig. 74, if the space traversed were a

vacuum, it would be quite immaterial whether W W were inches or millions of miles distant from S; the rays of heat would travel on until they were absorbed by the cold body which had attracted them. Thus the heat that we receive from the sun, is radiant heat that has travelled 95 millions of miles through space.

(278.) Radiant heat has the remarkable property of passing through *moderate* thicknesses of air or gas without appreciable loss, or heating the air sensibly, so that in ordinary cases we may admit that air and gases cannot be heated directly by radiant heat, but only by contact with heated bodies. With very great thicknesses, however, such as several miles, the loss of heat becomes manifest, as is shown by the diminished power of the sun at and near its rising and setting, when its rays, passing very obliquely through the atmosphere, traverse a much greater thickness of air than at mid-day. For the same reason, the force of solar radiation varies throughout the year with the varying altitude of the sun, as shown by cols. 2 and 3 of Table 96. Thus, a thermometer with a blackened bulb freely

TABLE 96.—Of the FORCE of RADIATION from the SUN, and to the SKY.

Months.	Average Maximum Temp. of Air in Shade.	Solar Radiation. Mid-day Temp. in Sun.		Average Minimum Temp. of the Air.	Sky Radiation. Nocturnal Temp. of Sky.		Daily Range.	
		Mean.	Max.		Mean.	Min.	Sun and Sky. Cols. 3, 6.	In Air. Cols. 1, 4.
Jan.	43°·2	47°	55°	33°·7	30°	24°	31°	9°·5
Feb.	44·7	55	81	34·2	30	24	57	10·5
March	50·0	66	99	35·3	30	25	76	14·7
April	56·8	85	104	38·6	32	26	78	18·2
May	64·4	95	121	44·2	40	29	92	20·2
June	71·2	111	136	50·2	45	33	103	21·0
July	73·8	110	129	53·2	50	38	91	20·6
Aug.	72·8	106	132	53·4	48	40	92	19·4
Sept.	67·4	99	121	48·9	44	37	84	18·5
Oct.	58·3	86	101	43·7	39	33	68	14·6
Nov.	49·3	56	73	37·7	34	28	45	11·6
Dec.	45·0	50	57	35·5	32	25	32	9·5
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)

exposed to sun and air (40) at mid-day, will show on an average, $47 - 43 \cdot 2$ = say 4° in January, and $111 - 71 \cdot 2$ = say 40° in June, as the excess above the ambient air due to solar radiation: the average *maximum* excess is $55 - 43 \cdot 2$ = say 12° in January, and $136 - 71 \cdot 2$ = say 65° in June.

Celestial space has a very low temperature (41), and although it is a vacuum, or nearly so, yet its infinite thickness enables it to become a powerful absorbent of heat, notwithstanding the counteracting effect of the heated earth and atmosphere. This is shown by cols. 5 and 6 of Table 96; the *mean* nocturnal depression of a thermometer exposed to the sky, but screened from the earth's radiation, being in January $33 \cdot 7 - 30 = 3° \cdot 7$, and in June $50 \cdot 2 - 45 = 5° \cdot 2$ below the ambient air. The average *maximum* depression with a serene and unclouded sky, is $33 \cdot 7 - 24$ = say 10° in January, and $50 \cdot 2 - 33$ = say 17° in June.

(279.) "*Loss of Heat by Contact of Air.*"—The loss of heat by contact of cold air is independent of the nature of the surface, so that cast iron, stone, wood, &c., &c., would lose the same amount of heat, under the same conditions of temperature, but the *form* of the body affects the result considerably, so that a plane, a sphere, and a cylinder will lose different amounts of heat per square foot in the same time.

Let P, in Fig. 75, be a vertical plane of *any* material, 1 foot square, having its surface S maintained at 60°, and let S′ and W W W have also the temperature of 60°, while the air in contact with S is at 59°. There will in this case be no loss of heat from S by radiation or conduction, because W and S′ are of the same temperature as S, but heat will be given out to the cold air; and experiment has shown that the amount for 1°, as in our case, is ·5945 unit per square foot per hour for a plane 1 foot high, but it will not be the same per square foot for any other height, for reasons that are given in (281).

(280.) For small differences of temperature (between the air and the body), say up to 30°, we may admit that the loss of heat is simply proportional to that difference (see Table 105), and the rule becomes

$$A' = A \times d;$$

in which d is the difference of temperature between the air and the body; A' = the loss of heat by contact of air, in units per square foot per hour; and A = the loss for 1° difference of temperature between the body, and the air in contact with it. This last will have a value, varying with the *form* of the body, as we shall see presently.

With great differences of temperature, Dulong has shown that the loss of heat increases in a much higher ratio than that difference, so that, for instance, when a body is 450° above the temperature of the air, the loss per degree is double the loss with small differences of 20 or 30 degrees (see (315) and Table 105).

(281.) If we investigate the loss of heat by a vertical plane, we shall be able to see the reason for the varying value of A. A vertical plane, 1 foot high, is found by experiment to lose ·5945 unit per square foot per hour, when heated 1° above the air in contact with it; but a high plane or wall loses less per square foot, for the following reason: Let Fig. 81 represent a plane 4 feet high, heated to 60°, while the air in contact with it is at 59°. The air in immediate contact with the wall being heated by it, is expanded thereby, becomes lighter than the surrounding air, and ascends in a constant current as shown by the arrows. Now for the first or lowest foot, the air is at 59° to begin with, but departs from it at a slightly increased temperature, so that there is *less* than 1° difference when it comes in contact with the second foot, and for that reason it receives less heat from it than it did from the lowest foot, and so throughout, each successive foot receives the air at progressively increased temperature and imparts to it less and less heat.

The loss by a vertical plane is given by the rule

$$A = \cdot 361 + \left(\cdot 233 \div \sqrt{H}\right),$$

in which A = the loss in units per square foot per hour for 1° difference in temperature, and H = the height of the plane or wall in feet. Table 97 has been calculated by this rule, and it shows that a wall 4 feet high, as in Fig. 81, loses ·4780 unit

per square foot per hour, and by subtraction and analysis we may find the loss by each successive foot; thus a wall 2 feet high loses per Table 97, ·5280 per foot, or ·528 × 2 = 1·056

TABLE 97.—Of the VALUE of A for VERTICAL PLANES.

Height in Feet.	Units per Square Foot for 1° Difference.	Height in Feet.	Units per Square Foot for 1° Difference.
1	·5945	20	·4133
2	·5280	30	·4037
3	·4962	40	·3980
4	·4780	60	·3924
5	·4655	80	·3871
10	·4350	100	·3843

unit; and as the first foot loses ·5945 unit, the second must lose 1·056 − ·5945 = ·4615 unit; similarly, the third foot will lose (·4962 × 3) − (·5945 + ·4615) = ·4326 unit, and thus we obtain the numbers given in Table 98.

TABLE 98.—Of the Loss of HEAT from CONTACT of COLD AIR by a WALL 4 feet high.

	Units per Square Foot per Hour for 1° Difference.
1st, or lowest foot	·5945
2nd foot	·4615
3rd "	·4326
4th "	·4234
Total	1·9120
Per square foot, mean	·4780

This explanation of the reason why a high wall does not lose the same amount of heat per square foot as a low one, will give some idea of the cause of the difference between bodies having the same area, but differing in form; those bodies lose most heat whose form allows the most free access and circulation of the air which carries off their heat.

(282.) For a horizontal cylinder, the rule becomes

$$A = \cdot 421 + (\cdot 307 \div r),$$

in which A = the loss in units per square foot of surface per hour for a difference in temperature between the body and the air of 1° and r = the radius of the cylinder in inches. Of course the cylinder is supposed to be of infinite length, so as to exclude the consideration of the loss by the *ends*. The second column in Table 99 is calculated by this rule.

TABLE 99.—Of the Loss of HEAT from CONTACT of AIR with HORIZONTAL CYLINDERS and SPHERES, per square foot per hour, for a difference in Temperature of 1° Fahr.

Diameter in Inches.	Value of A.		Diameter in Inches.	Value of A.	
	Horizontal Cylinder.	Sphere.		Horizontal Cylinder.	Sphere.
	units.	units.		units.	units.
2	·7280	1·4110	9	·4892	·5962
3	·6256	1·0618	10	·4824	·5729
4	·5745	·8872	12	·4722	·5380
5	·5440	·7824	18	·4551	·4798
6	·5230	·7126	24	·4466	·4507
7	·5087	·6627	36	·4381	·4219
8	·4978	·6253	48	·4338	·4070
(1)	(2)	(3)	(1)	(2)	(3)

(283.) For a sphere the rule becomes

$$A = \cdot 3634 + (1\cdot 0476 \div r),$$

in which A and r have the same signification as before, and from this we obtain col. 3 of Table 99.

(284.) For a vertical cylinder we have the rule

$$A = \left\{\cdot 726 + \frac{\cdot 2163}{\sqrt{r}}\right) \times \left(2\cdot 43 + \frac{5\cdot 49}{\sqrt{h}}\right\} \times \cdot 2044,$$

in which h = the height in inches, and the rest as before. Table 100 has been calculated by this rule.

(285.) "*Loss of Heat by Conduction.*"—Let P, Fig. 76, be a plate of building stone, 1 foot square, 1 inch thick, having its surface S maintained at 60°, and let the walls W W W and the air in contact with S be at 60° also, while the surface S′ is

TABLE 100.—Of the Loss of HEAT from CONTACT of AIR with VERTICAL CYLINDERS, in Units per square foot per hour, for a difference in Temperature of 1° Fahr.

Diameter in Inches.	Height of Cylinder in Feet.							
	1	3	5	10	20	30	40	50
2	·7733	·6474	·6046	·5654	·5363	·5237	·5164	·5113
4	·7213	·6038	·5639	·5273	·5002	·4885	·4816	·4769
8	·6846	·5731	·5353	·5005	·4747	·4636	·4571	·4526
12	·6683	·5594	·5224	·4886	·4634	·4526	·4462	·4418
18	·6550	·5483	·5121	·4788	·4542	·4446	·4374	·4330
24	·6471	·5416	·5058	·4730	·4487	·4382	·4320	·4278
36	·6377	·5338	·4985	·4662	·4422	·4318	·4225	·4183

at 59°. There will then be no loss of heat by radiation or contact of air, because W and the air are at the same temperature as S, but a certain amount of heat will be transmitted through the material from S to S′; and for stone 1 inch thick, the loss will be 13·7 units per square foot per hour for 1° difference, as in our case. This amount will vary very greatly with the nature of the material, we will call it C, and its value is given by Table 101. The amount of heat also varies directly as the difference of temperature of the two surfaces S and S′, and inversely as the thickness, and hence we have the rule

$$C' = C \times d \div E,$$

in which C′ = the loss by conduction in units per square foot per hour, C = the conducting power of the material, E = the thickness of the plate in inches, and d = the difference of temperature of S and S′. Thus a wall of stone 20 inches thick, having one surface at 70° and the other at 40°, taking the value of C at 13·7, will transmit $13{\cdot}7 \times (70 - 40) \div 20 = 20{\cdot}55$ units per square foot per hour.

(286.) Fig. 77 represents a case in which the surface S loses heat *simultaneously* by radiation, contact of air, and conduction. Here W, W, W, the air, and the surface S′ are all 1° lower than S, and the *total* loss is the sum of the separate losses in (275), (279), and (285), namely, $U = \overset{\text{R}}{\cdot 736} + \overset{\text{A}}{\cdot 5495} + \overset{\text{C}}{13{\cdot}7} = 14{\cdot}9855$ units per square foot per hour

TABLE 101.—Of the VALUE of C, or the CONDUCTING POWER of MATERIALS, being the quantity of Heat in Units transmitted per square foot per hour, by a Plate 1 inch thick, the two surfaces differing in Temperature 1°. From the Experiments of PÉCLET.

	Value of C.
Copper	515
Iron	233
Zinc	225
Lead	113
Marble, grey, fine-grained	28
„ white, coarse-grained	22·4
Stone, calcareous, fine	16·7
„ „ ordinary	13·68
Glass	6·6
Baked Clay, brickwork	4·83
Plaster, ordinary	3·86
Oak, transmission perpendicular to the fibres ..	1·70
Walnut „ „ „ ..	·83
Fir „ „ „ ..	·748
Fir „ parallel to the fibres	1·37
Walnut „ „ „	1·40
Gutta-percha	1·38
India-rubber	1·37
Brick-dust, sifted	1·33
Coke, pulverized	1·29
Cork	1·15
Chalk, in powder	·869
Charcoal of Wool, in powder	·636
Straw, chopped	·563
Coal, small, sifted	·547
Wood Ashes	·531
Mahogany Dust	·523
Canvas of Hemp, new	·418
Calico, new	·402
White Writing-paper	·346
Cotton Wool and Sheep Wool (any density)	·323
Eiderdown	·314
Blotting-paper, grey	·274

(287.) If W, S′, and the air were not all of one and the same temperature, the case would be modified. Let H, in Fig. 78, be a cube of new cast-iron 1 foot square, heated to 100°, and placed in a room whose walls, B, B, are maintained at 40°, and let the internal air be at 60°; considering only the loss by one vertical side of the cube, the loss by radiation with new cast-iron will be by Table 95, = ·648 for 1° difference, therefore in

our case $\cdot 648 \times (100 - 40) = 38\cdot 88$ units, and the loss by contact of air will be $\cdot 5945 \times (100 - 60) = 23\cdot 78$ units, so that the total loss is $38\cdot 88 + 23\cdot 78 = 62\cdot 66$ units per square foot per hour.

(288.) But sometimes a body may be losing heat from one source, and at the same time receiving it from another; thus in Fig. 79, we have a cube of stone, say 3 feet square, whose temperature is maintained at 50°, placed in a room whose walls are at 40°, while the air is at 60°. Here the loss of heat by radiation is $\cdot 736 \times (50 - 40) = 7\cdot 36$ units per square foot per hour; but by contact of air, heat will be *received*, not lost, the air being 10° warmer than the cube. By Table 97, a plane 3 feet high will lose or gain $\cdot 4962$ unit per square foot per hour for 1°, therefore in our case the heat received will be $\cdot 4962 \times (60 - 50) = 4\cdot 962$ units per square foot per hour, so that the final result is a loss of $7\cdot 36 - 4\cdot 962 = 2\cdot 398$ units per square foot per hour. Here then we have a case in which a body is *simultaneously losing and receiving heat* from different sources.

(289.) These rules and tables are easily applied to practice; thus a wall of stone, 20 feet high and 50 feet long at 70°, exposed to air and all surrounding objects at 60°, will lose $(\overset{R}{\cdot 736} + \overset{A}{\cdot 4133}) \times 20 \times 50 \times 10 = 11493$ units per hour; a horizontal cylinder, of rusted cast-iron, 6 inches diameter and 40 feet long, having an area of $3\cdot 14 \times \cdot 5 \times 40 = 62\cdot 8$ square feet, will lose, for say 15° difference $(\overset{R}{\cdot 6868} + \overset{A}{\cdot 523}) \times 62\cdot 8 \times 15 = 1140$ units per hour; and a sphere 2 feet diameter, having by the rules of mensuration an area of $3\cdot 14 \times 4 = 12\cdot 56$ square feet, say painted with oil paint, will lose for 15° difference $(\overset{R}{\cdot 7583} + \cdot 4507) \times 12\cdot 56 \times 15 = 228$ units per hour.

(290.) "*The Loss of Heat by Buildings.*"—Having thus determined the value of the three elements R, A, and C, we will proceed to apply them to buildings artificially heated for the purposes of life, which is one of the most important subjects to which they are applicable. The heat lost by buildings may be

divided into four portions; namely, that lost by the floor, by the ceiling, by the walls, and by the windows.

"*Loss by the Floor or Ground.*"—It is stated in (43) that the surface of the earth in ordinary cases varies in temperature with that of the air, but where the ground is covered by a building the case is different; it will then take pretty nearly the mean temperature of the earth at that place. We have shown (43) that at a depth of 20 feet the earth has a uniform and invariable temperature, having the mean yearly temperature of the air at that place; this in the case of London is shown by Table 34 to be about 50°, and as this is pretty nearly the average temperature of our dwellings in the cold season, the ground or floor, being in contact with earth beneath and air above, both of that temperature, will take the same, and the loss of heat from this source will be nothing.

"*Loss by the Ceiling, &c.*"—When the room is covered with an ordinary lath-and-plaster ceiling, and that again protected from cooling influences by the roof, the loss will be so small that it may be neglected, and we shall consider in what follows that the loss of heat from this cause is nothing. But where there is no ceiling, and the roof itself is exposed to the heated air in the building, there will be a great loss of heat, but one exceedingly difficult to estimate.

(291.) "*Loss of Heat by the Walls.*"—We will take for illustration the case of a room with brick walls and no windows, &c., as in Fig. 80, exposed on all sides to cooling influences, the external air and all the surrounding radiant objects having a temperature of 30°, while the internal air is maintained at 60°; and the problem to be solved is the amount of heat which will be received from the internal air, transmitted through the walls, and finally dissipated on the external objects.

To avoid complicating the question, we may assume that the room, Fig. 80, is of very large dimensions, so that the external surface has sensibly the same area as the internal surface. In practice this is near enough the truth in most cases, but it may perhaps be more correct to take the area at a mean between the two.

We are met at the outset with the difficulty that the tempera-

tures of the *surfaces* of the walls are not known; in our case (and in most cases) we only know the temperature of the internal and external *air*, T and T′; the interior surface of the wall t must obviously have a temperature lower than the internal air T, otherwise it could not receive heat from it: similarly the exterior surface t' must have a higher temperature than the external air, &c., or it could not give out heat to it.

(292.) We will first calculate the temperature of the two surfaces of the wall t and t', and shall adopt the following general notation throughout:

T = the temperature of the internal air.
T′ = „ external air.
t = „ internal surface of wall.
t' = „ external surface of wall.
t'' = „ glass in windows, &c.
R = radiant power of the material, as per Table 95.
A = loss by contact of air, as per Tables 97, 98, &c.
C = conducting power of the material, as per Table 101.
Q = R + A.
U = units of heat per hour.
E = thickness of the wall in inches.

(293.) Taking the walls in our case, Fig. 80, at 40 feet high, we have by Table 97 the value of A = ·398; the value of R for brickwork we shall take at ·736, and C at 4·83, therefore Q = 1·134, and we may find the temperature t of the internal surface of the wall by the formula

$$t = \frac{\left\{Q \times [E \times A \times T) + (C \times T']\right\} + \left\{A \times C \times T\right\}}{\left\{C \times [2 \times A) + R]\right\} + \left\{E \times A \times Q\right\}},$$

which, with a 14-inch wall, &c., as in our case, becomes

$$\frac{\left\{1\cdot134 \times [14 \times \cdot398 \times 60) + (4\cdot83 \times 30]\right\} + \left\{\cdot398 \times 4\cdot83 \times 60\right\}}{\left\{4\cdot83 \times [2 \times \cdot398) + \cdot736]\right\} + \left\{14 \times \cdot398 \times 1\cdot134\right\}} = 48°\cdot02$$

the internal temperature of the wall.

(294.) Having found t, we may find t' by the formula

$$t' = \frac{(C \times t) + (Q \times E \times T')}{C + (Q \times E)},$$

which in our case becomes

$$\frac{(4 \cdot 83 \times 48 \cdot 02) + (1 \cdot 134 \times 14 \times 30)}{4 \cdot 83 + (1 \cdot 134 \times 14)} = 34^\circ \cdot 2,$$

which is the temperature of the external surface of the wall.

Having thus found t and t', we may now calculate the quantity of heat transmitted, by several formulæ.

(295.) Knowing the temperature of the internal air T, and of the internal surface of the wall in contact with it t, we can easily calculate the amount of heat which the wall will absorb, namely $U = A \times (T - t)$, which in our case is $\cdot 398 \times (60 - 48 \cdot 02) = 4 \cdot 77$ units per square foot per hour.

(296.) We may also calculate from the known temperatures of the two surfaces of the wall t and t', and the conducting power of the material C, irrespective of the temperatures of the internal and external air, by the rule

$$U = \frac{C \times (t - t')}{E},$$

which in our case is $\dfrac{4 \cdot 83 \times (48 \cdot 02 - 34 \cdot 2)}{14} = 4 \cdot 77$ units per square foot per hour, as before.

(297.) We may also calculate from the known temperatures of the external surface of the wall t', and the temperature of the external air and surrounding objects which absorb radiant heat, both being at 30°, by the rule

$$U = Q \times (t' - T),$$

which in our case becomes $1 \cdot 134 \times (34^\circ \cdot 2 - 30^\circ) = 4 \cdot 77$ units, as before.

(298.) These three formulæ show the steps of the cooling process; (295) shows that $4 \cdot 77$ units are absorbed from the internal air by the internal surface of the wall; (296) shows that the same amount of heat is transmitted in the same time from one surface of the wall to the other; and (297) proves

that the heat thus received and transmitted to the outer surface, is finally dissipated by it on the external air, &c. It should be observed that the internal surface receives its heat only by contact of warm air, and none by radiation, because all the internal walls have one and the same temperature with the wall B, whose condition we have been examining; but the exterior surface loses its heat both by radiation and contact of air. The result of this is, that while the difference of temperature between the air and surface of wall is internally $60 - 48 \cdot 02 = 11^\circ \cdot 98$, externally it is only $34^\circ \cdot 2 - 30^\circ = 4^\circ \cdot 2$: this, however, varies with the thickness of the wall, as shown by cols. 3 and 4 in Table 102.

(299.) We may also determine the loss if we know only t and T′, or the temperature of the *interior* surface of the wall and of the *external air*, by the rule

$$U = \frac{Q \times (t - T')}{1 + \left(Q \times \frac{E}{C}\right)},$$

which in our case becomes $\dfrac{1 \cdot 134 \times (48 \cdot 02 - 30)}{1 + \left(1 \cdot 134 \times \frac{14}{4 \cdot 83}\right)} = 4 \cdot 77$ units, as before. This is a useful rule for many cases, such as the wall of a furnace which has the temperature of the fire itself on one side, and is exposed to the air and radiant objects at low temperatures on the other side.

(300.) The loss of heat may also be calculated from the known temperatures of the external and internal *air* only: this is, perhaps, the most useful formula, because ordinarily these are the only known temperatures; the rule then becomes

$$U = \frac{(A \times C \times Q) \times (T - T')}{\left\{C \times [2 \times A) + R]\right\} + \left\{E \times A \times Q\right\}},$$

which for our case becomes

$$\frac{(\cdot 398 \times 4 \cdot 83 \times 1 \cdot 134) \times (60 - 30)}{\left\{4 \cdot 83 \times [2 \times \cdot 398) + \cdot 736]\right\} + \left\{14 \times \cdot 398 \times 1 \cdot 134\right\}} = 4 \cdot 77 \text{ units},$$

as before.

(301.) Table 102 has been calculated by these rules for stone and brickwork, for the latter we have taken C at 4·83, and for stone at 13·7, while R and A have the same value for both, namely, R = ·736, and A = ·398. Thus, take the case of a building, say 40 feet high, 100 feet long, and 60 feet wide, exposing 12,800 square feet, with internal air at 60° and external air at 30°; each square foot of brickwork, say 14 inches thick, will lose by the table ·159 unit for 1°, or for 30° ·159 × 30 = 4·77 units per hour, and the total loss will therefore be 4·77 × 12800 = 61056 units. With stone walls 2 feet thick the loss would have been ·194 × 30 × 12800 = 74496 units. For walls a little more or less than 40 feet high, the loss will be very nearly the same as per Table 102, and for ordinary cases the tabular numbers are correct enough for practice.

TABLE 102.—Of the VALUE of U, or the LOSS of HEAT in UNITS per SQUARE FOOT per HOUR by a BUILDING exposed on all sides to AIR, &c., at 30°, the internal Air being 60°, Walls 40 feet high.

BRICKWORK.

Thickness.		Temperature of the Wall.		Value of U.	
Brick.	Inches.	Internal Surface.	External Surface.	For 30°. Units.	For 1°. Units.
½ =	4½	42°57	36°11	6·936	·2312
1 =	9	45·78	35·06	5·730	·1910
1½ =	14	48·02	34·20	4·770	·1590
2 =	18	49·4	33·71	4·212	·1404
3 =	27	51·6	32·94	3·339	·1113
4 =	36	53·0	32·43	2·76	·0920

STONE WALLS.

Inches.	Internal Surface.	External Surface.	For 30°. Units.	For 1°. Units.
6	40·33	36·90	7·83	·261
12	42·34	36·19	7·02	·234
18	43·96	35·61	6·36	·212
24	45·33	35·13	5·82	·194
30	46·49	34·73	5·37	·179
36	47·47	34·39	4·98	·166

(302.) We have so far considered only the case of a building exposed on *all sides* to cooling influences. When only one face

is exposed, as in the case of a room forming part of a large building, the case is somewhat different, and may be illustrated by Fig. 84, in which W is a room, only one wall of which is exposed to the external air and radiant objects at 30°, the internal air and surfaces of walls G, H, J, being maintained at 60°, and let F, F, F, &c., be other rooms. The interior surface of the wall B, whose condition we are now to consider, is therefore exposed not only to *air* at 60°, but also to radiation from the other walls at 60° also; it will therefore have a higher temperature than before, and will in consequence transmit more heat. Being exposed to similar influences on both sides, that is to say, being heated on one side by air and radiant bodies having one and the same temperature, and cooled on the other in the same manner, and the value of R and A being the same for both internal and external surfaces, the temperature of the wall *at the centre of its thickness* will be a mean between T and T′, or $(60 + 30) \div 2 = 45°$, as in Fig. 83, which represents a portion of the wall B on a larger scale.

It will be seen by inspecting the figures, that this is analogous to the case of a wall 7 inches thick, with one surface maintained at 45°, while the other is exposed to air and radiant objects at 30°, as at O P, and we can calculate the amount of heat transmitted by our formula (299), which in our case becomes $\dfrac{1 \cdot 134 \times (45 - 30)}{1 + \left(1 \cdot 134 \times \dfrac{7}{4 \cdot 83}\right)} = 6 \cdot 4$ units per square foot per hour. This is the amount transmitted from M to N, Fig. 83, and the same amount would pass from L to M because the conditions are the same, namely, the same thickness of wall, or 7 inches, and difference of temperature, or 15°; so that a wall 14 inches thick exposed to air and radiant objects at 60° on one side, and air and radiant objects on the other side at 30°, will transmit 6·4 units per square foot per hour, and this we have found without knowing the temperatures of the surfaces of the wall.

(303.) We can now find the temperatures of the two surfaces of the wall B; the internal surface has to receive 6·4 units, and as for 1° it would receive only 1·134 unit (namely, the value of Q), its temperature must be $6 \cdot 4 \div 1 \cdot 134 = 5° \cdot 64$ lower than

the internal air, &c., or $60° - 5°·64 = 54°·36$. Again, the external face has to lose 6·4 units, and must be 5°·64 warmer than the external air in order to do so, its temperature will therefore be $30° + 5°·64 = 35°·64$.

Knowing now the temperatures of the two faces of the wall B, its thickness, and conducting power C, we may verify these calculations by the formula in (296), which in our case becomes $\frac{4·83 \times (54°·36 - 35°·64)}{14} = 6·4$ units, as before. This is considerably more than the loss by a building exposed on all sides, which we found to be 4·77 units for the same internal and external temperature. Table 103 gives the loss in units per square foot per hour for buildings of brick and stone.

TABLE 103.—Of the Loss of HEAT per SQUARE FOOT per HOUR by BRICK and STONE WALLS, 40 feet high, in Buildings where only one face is exposed, and for 1° difference of Internal and External Temperature.

Brickwork.		Stone.	
Thickness.	U.	Thickness.	U.
brick. inches.		inches.	
½ = 4½	·371	6	·453
1 = 9	·275	12	·379
1½ = 14	·213	18	·324
2 = 18	·182	24	·284
3 = 27	·136	30	·257
4 = 36	·108	36	·228

(304.) It may appear anomalous that a building exposed on all sides should lose less heat per square foot than a wall in a room exposed to cooling influences on one side only. We assumed for our illustration in Fig. 84 that the temperature of the internal surfaces of the walls G, H, J, in the room W was 60°; but we have seen that the wall B borrows heat from them to the extent of $·736 \times (60° - 54°·36) = 4·15$ units per square foot per hour; H will furnish the largest share, and we may assume it at one-half, and as the areas of H and B are equal, each square foot of H will give out to B, $4·15 \div 2 = 2·075$ units per square foot per hour. But to do this, its surface K must be of

higher temperature than H, and the difference of temperature by the rule $\frac{U \times E}{C} = (t - t')$ will be in our case $\frac{2 \cdot 075 \times 9}{4 \cdot 83} = 3^\circ \cdot 86$; the temperature of K must therefore be $60^\circ + 3^\circ \cdot 86 = 63^\circ \cdot 86$: and as we may suppose that the heat received by K is derived from the air in the room F′, that air must be still warmer than K to the extent of $2 \cdot 075 \div \cdot 398 = 5^\circ \cdot 2$; the temperature of the air in F′ will therefore be $63^\circ \cdot 86 + 5^\circ \cdot 2 = 69^\circ \cdot 06$. Thus it appears that to secure the conditions we assumed for the room W, with air and walls at 60°, a higher temperature is necessary in the rest of the building, and for that reason more heat is lost by the wall B than in a case like Fig. 80.

(305.) "*Loss of Heat by Glass in Windows, &c.*"—We will first take the case of a window in the room, Fig. 85, in which the interior walls and internal air in contact with the glass have one and the same temperature of 60°, and all the external radiant objects and external air have one and the same temperature of 30°. In that case, the glass being heated on one side and cooled on the other by similar influences, will have a temperature in the centre of its thickness (302), a mean between the two, or in our case $t'' = (60 + 30) \div 2 = 45^\circ$, and with thin glass we may assume that it has this temperature throughout. We may calculate the amount of heat received from within and dissipated without, by the rule

$$U = Q \times (T - t'').$$

For glass the value of R is by Table 95, ·5948, and the value of A by Table 97 for a window say 5 feet high, is ·4655; therefore $Q = \cdot 5948 + \cdot 4655 = 1 \cdot 0603$, and in our case the loss is $1 \cdot 0603 \times (60^\circ - 45^\circ) = 15 \cdot 9$ units per square foot per hour for 30° difference of internal and external temperature, or $15 \cdot 9 \div 30 = \cdot 53$ unit for 1° by a window 5 feet high; for 10 feet and 20 feet the losses are ·515 and ·504 respectively.

(306.) "*Double Windows.*"—With a double window the loss of heat would be considerably less than with a single glass. This case is shown approximately by Fig. 86½; the temperature of the inside glass x is now 51° instead of 45°, that of the outside glass z being 39° The glass x will receive from the

internal air $\cdot 4655 \times (60 - 51) = 4 \cdot 19$ units, and from the inside walls by radiation $\cdot 5948 \times (60 - 51) = 5 \cdot 35$ units, or $9 \cdot 54$ units total. The air between the glasses will in this case have a temperature an arithmetical mean between the external and internal air, or 45°, and the glass z will receive from it $\cdot 4655 \times (45 - 39) = 2 \cdot 79$ units, and by radiation from x, $\cdot 5948 \times (51 - 39) = 7 \cdot 14$ units, or $9 \cdot 93$ units total, which is slightly in excess, but is sufficiently near for our purpose. Then the glass z will give out to the external air $\cdot 4655 \times (39 - 30) = 4 \cdot 19$ units, and to external radiant objects $\cdot 5948 \times (39 - 30) = 5 \cdot 35$ units, or total $9 \cdot 54$ units per square foot per hour, being the same as that received by x. The ratio of the losses by double and single windows is therefore $9 \cdot 54 \div 15 \cdot 9 = \cdot 6$ to 1. Péclet found by experiment that the loss varied slightly with the space between the glasses, possibly because larger spaces allow more free circulation of enclosed air; with $\cdot 8$ inch the ratio was $1 \cdot 7 \div 3 \cdot 66 = \cdot 47$ to 1; with 2 inches $2 \div 3 \cdot 66 = \cdot 55$ to 1; and at that rate, with $2 \cdot 8$ inches, which is perhaps the distance that would be adopted in practice, the ratio would be $\cdot 6$, as found by our calculation. The advantage of double windows over single ones is not only that they transmit less heat, but also that the temperature of the inside glass being greater, less *radiant* heat is absorbed from the occupants of the room (310).

(307.) We will now consider the loss of heat by a window in a room like Fig. 80, in which the glass is exposed internally to air at 60°, and radiant walls at 48°·02, while externally the temperature both of the air and radiant objects is at 30°. In this case the temperature of the glass will not be the arithmetical mean between T and T′, but will be given by the formula

$$t'' = \left(\frac{\mathrm{T} - t) \times \mathrm{A}}{\mathrm{A} + \mathrm{R}} + t + \mathrm{T}'\right) \div 2,$$

which in our case becomes

$$\left(\frac{60 - 48 \cdot 02) \times \cdot 4655}{\cdot 4655 + \cdot 5948} + 48 \cdot 02 + 30\right) \div 2 = 41^\circ \cdot 65,$$

the temperature of the glass. This case is shown by Fig. 82; the heat received from within by contact of the air will be,

$A \times (T - t'')$, or in our case, $\cdot 4655 \times (60^\circ - 41^\circ{\cdot}65) = 8{\cdot}541$; and by radiation, $= R \times (t - t'')$; in our case, $\cdot 5948 \times (48{\cdot}02 - 41{\cdot}65) = 3{\cdot}789$ units, making a total of $8{\cdot}541 + 3{\cdot}789 = 12{\cdot}3$ units per square foot per hour. It is obvious that the same amount of heat has to be dissipated on the external objects; here, however, the loss arises from radiation and contact of air both at the same temperature, and we have $U = (A + R) \times (t'' - T')$; in our case $(\cdot 4655 + \cdot 5948) \times (41^\circ{\cdot}65 - 30^\circ) = 12{\cdot}3$ units, or the same as the heat received, proving that the temperature we found for the glass is correct. For 1° difference of internal and external air, the loss is $12{\cdot}3 \div 30 = \cdot 41$ unit per square foot per hour for a window 5 feet high; for 10 feet high and 20 feet the losses are $\cdot 4$ and $\cdot 391$ respectively.

(308.) "*Loss by Glasshouses, Conservatories, &c.*"—We will lastly consider the case of a glasshouse exposed on all sides to air and radiant objects at 30°, the internal air being maintained at 60°, as in Fig. 86; here all the internal surfaces having the same temperature, all the heat received from the interior must be given out by the contact of heated air only; but the heat thus received is dissipated on external objects both by radiation and contact of air. The temperature of the glass will therefore not be a mean between T and T′, but will be found by the formula

$$t'' = \frac{(A \times T) + \left\{A + R) \times T'\right\}}{(2 \times A) + R}.$$

Taking the height at 10 feet, our Table 97 gives $\cdot 435$ for the value of A, and R being $\cdot 5948$ as before the rule becomes in our case

$$\frac{(\cdot 435 \times 60) + \left\{\cdot 435 + \cdot 5948) \times 30\right\}}{(2 \times \cdot 435) + \cdot 5948} = 38^\circ{\cdot}9,$$

which is the temperature of the glass: the heat received from within will be $A \times (T - t'')$; in our case, $\cdot 435 \times (60^\circ - 38^\circ{\cdot}9) = 9{\cdot}17$ units per square foot per hour; and the heat dissipated without will be $(A + R) \times (t'' - T')$, in our case $(\cdot 435 + \cdot 5948) \times (38^\circ{\cdot}9 - 30^\circ) = 9{\cdot}17$ units also. For 1° difference of internal

and external temperature we have therefore $9 \cdot 17 \div 30 = \cdot 306$ unit per square foot per hour.

(309.) Thus for the three cases we have investigated, we have found three different values of U under otherwise similar circumstances, namely, ·53 for a window in a room with only one face exposed; ·41 for a window in a brick building with 14-inch walls, exposed on all sides; and ·306 for a glasshouse. Comparing the loss by glass with that by the 14-inch walls, we find in the first case $\cdot 53 \div \cdot 213 = 2 \cdot 5$, and in the second $\cdot 41 \div \cdot 159 = 2 \cdot 58$ times more heat lost by the glass than by the walls.

(310.) The foregoing facts and results will enable us to explain some anomalies which are matters of common observation, but the philosophy of which is not generally understood. Thus, for instance, it is well known that a room warmed by an open fire is much more comfortable to the occupants than if heated by steam or hot-water pipes. Fig. 80 may illustrate this case; say that the air is heated by steam-pipes to 60°, and imparts its heat to the walls and the window, both are therefore of necessity colder than the air, and our figure shows that the walls are about 12°, and the glass 19°, colder than the air. The occupant is therefore suffering from the abstraction of his animal heat by radiation of walls at 48° and windows at 41°, while the air is at the comfortable temperature of 60°, and if we would raise the *walls* to 60°, it could only be done by raising the temperature of the air to about 75°, which would be oppressive and injurious.

(311.) When a room is warmed by an open fire, the walls are warmer than the air, for radiant heat has the remarkable property of passing through air (in moderate thickness) without raising its temperature (278). By a common open fire, radiant heat only is given out usefully, for all the heated air passes up the chimney and is practically lost, and the radiant heat, passing through the air without sensible loss, is absorbed by the walls and raises their temperature. The air enters the room by the crevices in the door-way and windows, at the external temperature, and is heated by contact with the walls, but must obviously be always colder than the walls by which it is heated, so that the occupant breathes air refreshingly cool, while the walls

being comparatively highly heated, do not absorb his animal heat with inconvenient rapidity.

It will be evident from this, that to obtain a comfortable temperature the walls themselves must be heated rather than the air. This may be done by causing hot air to pass by channels formed in the walls, see (374) and Fig. 110. The same result may be obtained by open fires or stoves highly heated, see (248).

(312.) "*Effect of Covers on Cooling.*"—When a heated body such as a vessel of hot water is covered by a metallic cover, or by a series of such covers, with air-spaces between each, but closed so that *that* air is not renewed, the rate of cooling follows a law which may be stated thus:—For any number n of enveloping covers the rate of cooling is equal to the rate of the vessel alone, freely exposed to the air, multiplied by the product of the surfaces of all the envelopes, and divided by the sum of all the possible products of $n - 1$ of the surfaces of the vessel and the envelopes. Thus, say we have a vessel and three covers, the ratio of whose four surface areas is 1, 2, 3, 4 respectively; we may make four different products of any *three* of those numbers: thus, taking away the first (1), the product of the other three is $2 \times 3 \times 4 = 24$; taking away the second (2), we obtain $1 \times 3 \times 4 = 12$; taking away the third (3), we have $1 \times 2 \times 4 = 8$; and finally, taking away the fourth (4), we have $1 \times 2 \times 3 = 6$, and no further combination of three is possible. The sum of these four products is $24 + 12 + 8 + 6 = 50$, and the product of the three *covers* being $2 \times 3 \times 4 = 24$, we have $24 \div 50 = \cdot 48$ as the ratio of the loss of heat with three covers, that by the vessel alone being $1 \cdot 0$. Thus we have the rules:

With a single cover,

$$R = B \div (A + B).$$

With two covers,

$$R = (B \times C) \div \Big((B \times C) + (A \times C) + (A \times B)\Big).$$

With three covers,

$$R = (B \times C \times D) \div \Big((B \times C \times D) + (A \times C \times D) + (A \times B \times D) + (A \times B \times C)\Big).$$

in which R = the ratio of the loss of heat with the covers, that by the vessel alone being 1·0; A = the area of the surface of the heated vessel, and B, C, D that of the covers in the order of their sizes and distances from the vessel.

With the surfaces A, B, C, D in the ratios 1, 2, 3, 4, with one cover we have $R = 2 \div (1 + 2) = \cdot 667$; with two covers $(2 \times 3) \div \left(2 \times 3) + (1 \times 3) + (1 \times 2\right) = \cdot 545$; and with three covers,

$$R = (2 \times 3 \times 4) \div \left(2 \times 3 \times 4) + (1 \times 3 \times 4) + (1 \times 2 \times 4) + (1 \times 2 \times 3\right\} = \cdot 48.$$

Thus with 0, 1, 2, 3 covers the ratios are 1, ·667, ·545, ·48 respectively.

The ratio of the areas of the covers to that of the vessel and to one another is very influential on the result, small covers being the most effective. Thus with areas 1·0, 1·1, 1·2, 1·3 R = 1·0, ·524, ·3646, ·2905; with areas 1, 5, 6, 7, R = 1·0 ·833, ·732, ·6625; and with very large covers having areas of 1, 5, 25, 125, R = 1·0, ·833, ·8065, ·8013 respectively.

Péclet made experiments on this subject, the heated vessel being cylindrical terminated by two conical ends; it was of tin plate, and the enveloping covers were of the same form and material, with spaces of $\frac{2}{10}$ inch between each. The surfaces of the vessel and four covers were in the ratio 260, 320, 420, 480, and 560; in the same circumstances, the rates of cooling with the vessel naked, and successively covered with 0, 1, 2, 3, 4 covers, were in the ratio 1·0, ·60, ·43, ·36, ·30; by the rule we obtain 1·0, ·552, ·4113, ·3363, ·28 respectively.

The rules suppose that the covers are of the same form and same kind of radiating surface as the vessel. When this is not the case the question becomes very complicated. Péclet found with the same vessel as before, but coated with black varnish, and tin-plate covers as before, R = 1·0, ·38, ·35, ·30, ·25; and when *sheet-iron* covers were substituted, R became 1·0, ·59, ·44, ·34, ·31 respectively. Bell-*glass* covers with $\frac{2}{10}$ inch air-spaces gave R = 1·0, ·50, ·41, ·34, ·30 respectively; with the two last alone, R = ·42.

CHAPTER XIII.

LAWS OF COOLING AT HIGH TEMPERATURES.

(313.) With high temperatures and great differences of temperature, the simple formulæ we have given require correction, as we have stated in (276) and (280).

"*Loss by Radiation.*"—The rule in (276) assumes that the loss by radiation is simply proportional to the *difference* of temperature of the radiant body and the absorbent: but Dulong has shown that with the same difference the loss varies with the *absolute temperature* of the absorbent, so that, for instance, if in Fig. 74 the temperature of W W had been 212° and of S = 213°, the loss of heat per degree would have been about double the amount with the respective temperatures 60° and 59°. The loss of heat increases also in a much more rapid ratio than the difference of temperature, thus, with 432° difference, and with the absorbent at 212° (the radiant being in that case at 644°), the loss per degree is six times greater than at low temperatures as in (276). Dulong has given rules which agree well with experiment, up to a difference in temperature of 468°. This rule is a very difficult one to apply, but it may be put in such a form as to give a *ratio* by which calculations by the simple rule may be easily corrected. The rule then becomes

$$\frac{124 \cdot 72 \times 1 \cdot 0077^{t} \times (1 \cdot 0077^{T} - 1)}{T} = R'',$$

in which t = the temperature of the absorbent, or recipient of radiant heat in degrees Centigrade; T = the *excess* of temperature of the radiating body in degrees Centigrade; and R″ = the *ratio* of loss of heat under the given temperatures. Table 104 has been calculated by this rule; but the temperatures are reduced to Fahrenheit's scale. The constant 124·72 is given by Péclet, who found the rule to agree perfectly with his own experiments. The loss of heat at *very* high temperatures, as calculated by this rule, becomes exceedingly great, as is shown by Table 106. In all probability there is considerable error in

TABLE 104.—For RADIANT HEAT; being the *ratio* of HEAT emitted or absorbed at DIFFERENT TEMPERATURES, according to the Experiments and Formula of DULONG.

Excess of Temperature of the Radiant above that of the Recipient in degrees Fahr.	Temperature of Recipient of Radiant Heat, in Degrees Fahr.											
	32°	50°	59°	68°	86°	104°	122°	140°	158°	176°	194°	212°
	Ratio of Heat Emitted or Absorbed.											
18°	·997	1·075	1·120	1·165	1·254	1·355	1·467	1·580	1·702	1·848	1·994	2·150
36	1·032	1·083	1·160	1·206	4·299	1·403	1·520	1·680	1·763	1·914	2·064	2·227
54	1·071	1·155	1·200	1·251	1·348	1·449	1·576	1·70	1·83	1·985	2·142	2·310
72	1·115	1·202	1·250	1·302	1·403	1·515	1·640	1·760	1·900	2·066	2·230	2·400
90	1·163	1·254	1·310	1·360	1·463	1·580	1·710	1·840	1·980	2·154	2·330	2·510
108	1·212	1·307	1·360	1·416	1·525	1·648	1·784	1·920	2·070	2·280	2·423	2·615
126	1·26	1·36	1·42	1·48	1·59	1·72	1·86	2·00	2·16	2·34	2·52	2·72
144	1·32	1·42	1·48	1·54	1·65	1·79	1·94	2·08	2·24	2·44	2·64	2·83
162	1·37	1·48	1·54	1·60	1·73	1·86	2·02	2·17	2·34	2·54	2·74	2·96
180	1·44	1·55	1·61	1·68	1·81	1·95	2·11	2·27	2·46	2·66	2·87	3·10
198	1·50	1·62	1·69	1·75	1·89	2·04	2·21	2·38	2·56	2·78	3·00	3·24
216	1·58	1·69	1·76	1·83	1·97	2·13	2·32	2·48	2·68	2·91	3·13	3·38
234	1·64	1·77	1·84	1·90	2·06	2·28	2·43	2·52	2·80	3·03	3·28	3·46
252	1·71	1·85	1·92	2·00	2·15	2·33	2·52	2·71	2·92	3·18	3·43	3·70
270	1·79	1·93	2·01	2·09	2·22	2·44	2·64	2·84	3·06	3·32	3·58	3·87
288	1·89	2·03	2·12	2·20	2·37	2·56	2·78	2·99	3·22	3·50	3·77	4·07
306	1·98	2·13	2·22	2·31	2·49	2·69	2·90	3·12	3·37	3·66	3·95	4·26
324	2·07	2·23	2·33	2·42	2·62	2·81	3·04	3·28	3·53	3·84	4·14	4·46
342	2·17	2·34	2·44	2·54	2·73	2·95	3·19	3·44	3·70	4·02	4·34	4·68
360	2·27	2·45	2·56	2·66	2·86	3·09	3·35	3·60	3·88	4·22	4·55	4·91
378	2·39	2·57	2·68	2·79	3·00	3·24	3·51	3·78	4·08	4·42	4·77	5·15
396	2·50	2·70	2·81	3·93	3·15	3·40	3·68	3·97	4·28	4·64	5·01	5·40
414	2·63	2·84	2·95	3·07	3·31	3·51	3·87	4·12	4·48	4·87	5·26	5·67
432	2·76	2·98	3·10	3·23	3·47	3·76	4·10	4·32	4·61	5·12	5·33	6·04

applying the rule to such extreme cases; but we obtain by it a nearer approximation to the truth than we could get without its assistance. This table shows that with a radiant body at a clear red heat of 1860°, the loss is about 300! times the amount due by the simple formula, and at a bright white heat of 2580°, it rises to 4604!! times that amount. This may be very incorrect; but the extreme rapidity with which a body at white heat cools down to orange and cherry red, &c., seems to indicate that at extreme temperatures the loss of heat is exceedingly rapid.

(314.) The application of Table 104 is very simple. Say we have a mass of wrought iron heated to 600° in a chamber whose walls are at 190°. By Table 95, the radiant power of an ordinary surface of wrought iron is ·5662, and by the simple rule we have $\cdot 5662 \times (600 - 190) = 232$ units per square foot per hour; but by Table 104, the nearest number to the temperature of the absorbent is 194°, and to the difference (or $600 - 190 = 410°$), is 414°, and the ratio for those two numbers is by the Table $= 5 \cdot 26$, and the true loss by radiation in our case is therefore $232 \times 5 \cdot 26 = 1220$ units per square foot per hour.

(315.) "*Loss by Contact of Air.*"—The researches of Dulong show that the loss by contact of cold air is independent of the *absolute* temperature of the heated body, differing in this respect from radiant heat; but he found that the heat lost increases more rapidly than the simple ratio of excess of temperature. Putting his formula, with the constants given by the experiments of Péclet, in such a form as to give us a *ratio* for the different temperatures, we have the rule

$$R''' = \frac{\cdot 552 \times t^{1 \cdot 233}}{t}$$

in which t = the difference of temperature of the body and the air in contact with it in degrees Centigrade, and R''' = the ratio of loss of heat with that difference; and thus we obtain the numbers in Table 105 and in col. 5 of Table 106. It will be observed that the departures from the simple law are much less than with radiant heat; at the extreme temperature of 2580°, and with air at 60°, the loss is only $2 \cdot 985$ times the amount given by the simple rule.

Applying this to the case in (314), say that the heated body was a vertical plane 4 feet high, Table 97 gives ·478 for the

TABLE 105.—Of the *Ratio* of HEAT EMITTED or ABSORBED by CONTACT of AIR with given DIFFERENCES OF TEMPERATURE.

Difference of Temperature of the Air and the Body in Contact.	Ratio of Heat.	Difference of Temperature of the Air and the Body in Contact.	Ratio of Heat.
18°	·94	252°	1·742
36	1·11	270	1·774
54	1·22	288	1·800
72	1·305	306	1·827
90	1·372	324	1·852
108	1·433	342	1·874
126	1·486	360	1·897
144	1·533	378	1·920
162	1·575	396	1·940
180	1·615	414	1·960
198	1·650	432	1·980
216	1·684	450	2·000
234	1·720	468	2·017

TABLE 106.—Of the RATIO of LOSS of HEAT at VERY HIGH TEMPERATURES, by the Formulæ of DULONG.

Temperature of the Heated Body.	Temperature of the Air in Contact with the Body, and of all Surrounding Objects.	Difference of Temperature of the Body, and of the Air, and Surrounding Objects.	Ratio of Heat lost at different Temperatures by	
			Radiation.	Contact of Air.
480°	60°	430°	3·10	1·980
600	”	540	4·19	2·085
780	”	720	7·17	2·230
960 Red, just visible ..	”	900	12·68	2·348
1140	”	1080	23·01	2·450
1320 Dull red	”	1260	42·70	2·540
1500 Dull cherry red ..	”	1440	80·67	2·620
1680 Cherry red	”	1620	154·5	2·693
1860 Clear red	”	1800	299·7	2·760
2220 Clear orange	”	2160	1159·0	2·880
2580 White, bright ..	”	2520	4604·0	2·985

NOTE.—For the loss at lower temperatures, see Tables 104 and 105.

value of A, and with the air at 190°, we have by the simple rule $\cdot 478 \times (600 - 190) = 196$ units; but for a difference of 410°, the nearest number in Table 105 is 414°, for which the ratio is 1·96, and hence we have $196 \times 1 \cdot 96 = 384$ units per square foot per hour. Adding the respective losses by radiation and contact of air together, we obtain $1220 + 384 = 1604$ units as the total loss per square foot per hour; see (247) and Table 88.

(316.) "*Steam-pipes.*"—We may apply these rules to the case of a steam-pipe. Say the air of the room and the walls are at 60°, and the steam-pipe at 210°, or 150° difference. Then the correction for radiant heat by Table 104 is say 1·5, and for contact of air by Table 105 = 1·55. We may take R or the radiant power of cast iron from Table 95 at ·7. Table 99 gives for horizontal cylinders of 2, 3, 4, and 6 inches diameter, the respective values of A at ·728, ·6256, ·5745, and ·523, and we have

	R. Diff. Ratio.	By Radiation.	A. Diff. Ratio.	By Contact.	Units per Sq. Ft. per Hour
For 2 in. diam.	(·7 × 150 × 1·5)	= 157·5, and	(·728 × 150 × 1·55)	= 169·5	= 327
3 "	"	"	(·6256 × 150 × 1·55)	= 145·5	= 303
4 "	"	"	(·5745 × 150 × 1·55)	= 133·5	= 291
6 "	"	"	(·523 × 150 × 1·55)	= 121·5	= 279

The weight of steam condensed to water at 212° per hour will be found by dividing the units of heat by 966 (the latent heat of steam), and we thus obtain ·338, ·312, ·301, and ·289 lb. of water per square foot per hour respectively.

(317.) "*Enclosed Pipes, &c.*"—It should be observed that these calculations apply strictly to the case of a pipe freely exposed to air and radiant walls, &c., both of the same temperature, and this is nearly true where pipes are fixed in the room to be heated; but where they are enclosed in small channels under the floor, the case is very different, for in that case the walls enclosing the pipe become highly heated, and were they not continuously cooled by the air passing through them, they would soon take the same temperature as the pipe itself, radiation from the pipe would cease altogether, and it would give out only about half the amount of heat. Taking the

4-inch pipe as an example from the cases just calculated, the loss by radiation is $\cdot 7 \times 150 \times 1\cdot 5 = 157\cdot 5$ units, or rather more than half of the total heat emitted, which was 291 units. But the heat received by radiation from the pipe is given out again by the wall to the air, and the temperature of the wall rises only until the two are equal to one another. This temperature we find by trial to be about 158°. The wall will then be $210 - 158 = 52°$ colder than the pipe, and the ratio by Table 104 being $1\cdot 83$ for a recipient at 158°, and the absorbent powers of brickwork being by Table 95 $\cdot 736$, we have $\cdot 736 \times 52 \times 1\cdot 83 = 69\cdot 7$ units received from the pipe. The amount given out by the wall to the air may be found by taking the value of A for a wall say 2 feet high from Table 97 at $\cdot 528$ and the ratio from Table 105 for $158° - 60° = 98°$ difference, at say $1\cdot 39$, and we have $\cdot 528 \times 98° \times 1\cdot 39 = 71\cdot 9$ units, or nearly the same amount as was received, showing that the temperature of 158° is nearly correct. The pipe at 210°, exposed to absorbing walls at 158°, and air at 60°, will lose $(\cdot 7 \times 52 \times 1\cdot 83) + (\cdot 5745 \times 150 \times 1\cdot 55) = 200\cdot 2$ units instead of 291 units, as we found for the case of air and walls of the same temperature; the ratio is $200\cdot 2 \div 291 = \cdot 688$, or say 70 per cent.; see also (409).

(318.) "*Effect of Polished Metal Surfaces, &c.*"—The amount of heat lost by contact of air is not affected by the nature of the surface of the body, but that lost by radiation varies exceedingly, and the sum total is modified considerably. This may be illustrated by taking the case of a horizontal pipe 4 inches diameter outside, as in the last examples, heated at 210°, with air at 60°, &c., but with varying character of radiating surface. Taking the values of R from Table 95, and A as before from Table 99, at $\cdot 5745$, we have the results shown by Table 107.

It will be observed, that the difference between a pipe in its ordinary state and a whitewashed one is very small indeed; blackening with lampblack increases the loss about 11 per cent.; and a blackened pipe loses $317\cdot 97 \div 139\cdot 55 = 2\cdot 27$ times as much heat as one of silvered copper highly polished like a mirror. When it is desired to lose as little heat as possible,

TABLE 107.—Of the Loss of HEAT by a HORIZONTAL PIPE 4 inches outside Diameter, heated to 210° and exposed to Air, &c., at 60°, showing the effect of different kinds of Radiating Surface.

Kind of Surface.	R.	Diff.	Ratio.	By Radiation. Units.	A.	Diff.	Ratio.	By Contact of Air. Units.	Total per Hour per Square Foot. Units.	Ratio, Cast Iron, rusted = 1.
Silvered Copper, polished ..	(·02657 ×	150 ×	1·5) =	5·98 +	(·5745 ×	150 ×	1·55) =	133·57	139·55	·4860
Tinned Iron do. ..	(·08585 ×	do.	) =	19·32 +		do.		do.	152·89	·5325
Iron, Sheet do. ..	(·092 ×	do.	) =	20·7 +		do.		do.	154·27	·5373
Whitewashed	(·6786 ×	do.	) =	152·7 +		do.		do.	286·27	·9971
Cast Iron, rusted	(·6868 ×	do.	) =	154·53 +		do.		do.	287·10	1·0000
Lampblacked	(·8196 ×	do.	) =	184·4 +		do.		do.	317·97	1·1070

NOTE.—The loss by pipes of other sizes, and by bodies of other forms will not differ very materially from the numbers given for a 4-inch horizontal pipe; so that this table may be used with approximate correctness for other cases. The difference between a whitened and a blackened pipe is shown to be less than is commonly supposed; the table gives the loss of a whitewashed pipe at $\frac{286 \cdot 27}{317 \cdot 97} = \cdot 9$, or 90 per cent. of the amount lost by a lampblacked one.

tinned iron (or common tin-plate) is a very effective and cheap material, losing less than half the amount of a blackened surface.

(319.) "*Effects of Thickness of Metal in Pipes heated internally.*"—We have assumed in the preceding calculations, and elsewhere throughout the work, that the outside of a steam-pipe has the same temperature as the steam, &c., inside, and this is practically true with thin pipes, such as are commonly used. With great thicknesses, however, the external temperature becomes sensibly less than that of the internal steam; it will be interesting to investigate the case generally.

We cannot calculate the loss of heat by the ordinary rules (285), because the surface dissipating the heat has a greater area than the surface receiving it; the ordinary rule supposes the wall, &c., transmitting heat to have parallel plain surfaces and to have a very large (or infinite) area, so that the heat may be considered as travelling in parallel lines from surface to surface, but in a thick pipe the heat travels in radial lines, and the amount transmitted cannot be calculated in the common way.

Let r = the radius of the inside of a pipe, in inches.
r' = " outside "
R = the radiant power of the outside surface, Table 95.
A = the loss of horizontal cylinder by contact of air (282), and Table 99.
Q = R + A.
C = conducting power of the material of the pipe, Table 101.
N = (log. r' − log. r) × 2·3.
t = temperature of the steam, &c., and of inside surface.
T′ = " external air, and radiant objects.
U = units of heat lost per square foot per hour.
U′ = " per foot run per hour.

Then we have the rule

$$U' = \frac{\cdot 5233 \times Q \times r' \times C \times (t - T')}{C + (Q \times r' \times N)}.$$

Taking the case of a pipe 4 inches bore, 4 inches thick in cast iron, 12 inches outside diameter, with steam at 212° and air, &c., at 60° or 152° difference, we may take R at ·7, A at

·4722, Q at 1·1722, and C at 233; for N we have log. of 6 = ·778, and log. of 2 = ·301, and N = (·778 − ·301) × 2·3 = 1·097, and the rule becomes $\frac{\cdot 5233 \times 1\cdot 1722 \times 6 \times 233 \times 152}{233 + (1\cdot 1722 \times 6 \times 1\cdot 097)}$ = 541·5 units per foot run; and a pipe 12 inches or 1 foot diameter, having an area of 3·14 square feet per foot in length, this is equal to 541·5 ÷ 3·14 = 172·4 units per square foot per hour.

(320.) But with a good conductor of heat, such as cast iron, the external temperature of the pipe will be high, and these numbers will require correction (313) (315) by Tables 104 and 105, and to apply these we require to know the temperature of the external surface.

By the rule in (297) U = Q × (t − T′), and hence U ÷ Q = (t − T′), which in our case becomes 172·4 ÷ 1·1722 = 147° above the atmosphere; the temperature of the outer surface must therefore be 147 + 60 = 207°, or 5° less than that of the inside of the pipe. With that temperature, the correction by Table 104 is 1·48, and for A by Table 105, 1·535, and calculating the true loss as in (316) we have (·7 × 1·48) + (·4722 × 1·535) × 147 = 258·8 units per square foot per hour, or 258·8 × 3·14 = 812·6 units per foot run.

With a pipe infinitely thin in metal: A becomes ·5745, Q 1·2745, C is cut out, and N being 0, the expression (Q × r' × N) vanishes, and the rule becomes ·5233 × Q × r' × (t − T′), or in our case ·5233 × 1·2745 × 152 = 202·77 units per foot run, and the area of a 4-inch being 3·14 × 4 ÷ 12 = 1·0467 square foot, the loss is 202·77 ÷ 1·0467 = 193·7 units per square foot.

This particular case might have been calculated by the common rule, Q × (t − T′) = U, or 1·2475 × 152 = 193·7 units per square foot per hour; the same as by the other rule.

Correcting these numbers by Tables 104 and 105, we have the true loss for 152° difference = (·7 × 1·5) + (·5745 × 1·55) × 152 = 294·9 units per square foot, or 294·9 × 1·0467 = 308·7 units per foot run per hour; so that the pipe 4 inches thick loses 812·6 ÷ 308·7 = 2·6 times as much heat per foot run, as one infinitely thin.

Table 108 has been calculated in this way; we find that even with the great thickness of 4 inches, the external temperature is only 5° less than that of the steam; with a thickness of 1 inch the difference is less than 1°, and of course with a pipe of the ordinary thickness of $\frac{5}{16}$ to $\frac{1}{2}$ inch, the external temperature will be sensibly the same as the internal, and the assumption that the heat lost is proportional to the external diameter is practically correct.

TABLE 108.—Of the Loss of HEAT by a HORIZONTAL CAST-IRON PIPE, 4 inches bore, with different Thicknesses of Metal, heated with Steam inside at 212°, and freely exposed to Air, &c., at 60°.

Diameter of the Pipe in Inches.		Thickness of Metal in Inches.	Loss of Heat per Hour.			Temperature.		
			Per Foot run.		Per Square Foot. Units.	Of the External Surface.	Of the Internal Surface.	Difference.
Inside.	Outside.		Units.	Ratio.				
						°	°	°
4	4	0	308·7	00	294·9	212	212	·000
4	6	1	441·17	1·4291	281·0	211·022	212	·978
4	8	2	571·4	1·851	273·0	210	212	2·0
4	12	4	812·6	2·600	258·8	207	212	5·0
4	16	6	1021·6	3·300	244·0	191	212	21·0

(321.) "*Steam-pipes, &c., cased in Bad Conductors of Heat.*" —The loss of heat by naked pipes to steam-engines, &c., is very considerable, and where the length is great it becomes serious, not only from the waste of fuel, but from the formation of water by condensation, which is obstructive to the working of the engine (126). Thus, with 35 lbs. steam, having by Table 71 a temperature of about 280°, the loss of heat by Table 90 with a 4-inch pipe would be 587 units per foot run per hour, and as by (118) a horse-power requires about 70,000 units, we find that a 4-inch steam-pipe, $4\frac{7}{8}$ diameter outside, and 100 feet long, would lose $(587 \times 100) \div 70000 = \cdot 84$ horse-power. This loss may be greatly reduced by casing the pipes in a material which conducts heat badly: see Table 101. The woollen felt which is made for this special purpose is the best and cheapest material.

Adopting the same notation as before (319), but putting C for the conducting power of the casing, and r'' for its outer radius, N will be $(\log. r'' - \log. r') \times 2 \cdot 3$, and the rule becomes

$$\frac{\cdot 5233 \times Q \times r'' \times C \times (t - T')}{C + (Q \times r'' \times N)} = U'.$$

For illustration of the effect of casing with different materials, we will take the case of a pipe 4 inches outside diameter, heated to 212°, the casing being covered in all cases with canvas, so as to give the same radiating power to the outer surface, and thus exhibit the variation in loss of heat due to conducting power alone.

Thus, for instance, with a casing of fir-wood, 1 inch thick, the outer diameter becomes 6 inches, $r'' = 3$ inches, R by Table 95 for canvas or calico $= \cdot 7461$, A by Table 99 $= \cdot 523$, therefore, $Q = 1 \cdot 2691$, and $C = \cdot 748$ by Table 101. For N we have log. $3 = \cdot 477$ and log. $2 = \cdot 301$, and $N = (\cdot 477 - \cdot 301) \times 2 \cdot 3 = \cdot 405$. Then the rule becomes

$$\frac{\cdot 5233 \times 1 \cdot 2691 \times 3 \times \cdot 748 \times 152}{\cdot 748 + (1 \cdot 2691 \times 3 \times \cdot 405)} = 98 \cdot 9,$$

or say 99 units per foot run per hour.

Calculating in this way with the different conducting powers and thicknesses, we obtain the numbers in Table 109.

TABLE 109.

	Thickness of Casing in Inches.					
	C.	½ inch.	1 inch.	2 inches.	4 inches.	6 inches.
Woollen Felt, or Cotton Wool	·323	79·5	52·4	34	22·5	18·1
Sawdust	·523	108	76·7	52	35·6	28·92
Fir-wood (transmission perpendicular)	·748	131	99	71	49·6	40·7
Coal Ashes (Coke pulverized)	1·29	165	138	108	80·6	67·6
Plaster	3·86	216	216	209	189·0	172·0
Stone	13·68	244	272	316	367·0	388·0
Marble, grey, fine ..	28·00	250	287	352	452·0	519·0

(322.) With low conductors, such as woollen felt, and considerable thicknesses, such as 4 or 6 inches, these numbers are correct enough for practice, but with thin casings, and better conductors, the external temperature will be high, and they require correction by Tables 104 and 105. We must first find the temperature: continuing the case with fir-wood, in which we have a loss of 99 units per foot run, the area of a 6-inch cylinder is $3 \cdot 14 \times 6 \div 12 = 1 \cdot 57$ square foot, and the loss of heat per square foot is $99 \div 1 \cdot 57 = 63$ units, and the excess of temperature being $U \div Q = (t - T')$, becomes in our case $63 \div 1 \cdot 2691 = 50°$ above the external air, and the temperature of the external surface is $50 + 60 = 110°$. With 50° excess, the correction of R by Table 104 is 1·2, and of A by Table 105 = 1·22, and the true loss of heat is $(\cdot 7461 \times 1 \cdot 2) + (\cdot 523 \times 1 \cdot 22) \times 50 = 76$ units per square foot, or $76 \times 1 \cdot 57 = 119$ units per foot run per hour.

Comparing this with the loss by a 4-inch pipe uncased, but still covered with thin canvas, the corrections for 152° difference by Tables 104 and 105 are respectively 1·5 and 1·55 as in (316), and we have $(\cdot 7461 \times 1 \cdot 5) + (\cdot 5745 \times 1 \cdot 55) \times 152 = 305 \cdot 4$ units per square foot, or $305 \cdot 4 \times 1 \cdot 0467 = 320$ units per foot run, the ratio is $119 \div 320 = \cdot 372$, the uncased pipe being 1·0.

Table 110 has been calculated in this way throughout: it will be observed that with low conductors, such as woollen felt, the loss becomes rapidly less with increase in thickness, but with such good conductors as stone and marble, the loss with all thicknesses is *greater* than by an uncased pipe, and the thicker the casing, the greater the loss; it is therefore worse than useless to use even moderately good conductors for such a purpose, and it is to illustrate this fact that such materials are introduced in the table.

(323.) The effect of a given thickness of casing is not quite the same for all diameters of pipe; Table 111 gives the loss of heat by horizontal pipes of different diameters from 2 to 12 inches, cased with from ¼ inch to 6 inches of felt, &c. The low temperature of the external surface with considerable thicknesses

TABLE 110.—Of the Loss of HEAT by a HORIZONTAL PIPE 4 inches OUTSIDE cased in different Thicknesses of various Materials and covered with of the Casing on

Material of the Casing.	Conducting Power of the Casing (C).	½ Inch thick. Loss of Heat per Hour. Per Foot run. Units.	Ratio.	Per Foot square. Units.	Temp. of the External Surface.	1 Inch thick. Loss of Heat per Hour. Per Foot run. Units.	Ratio.	Per Foot square. Units.	Temp. of the External Surface.
Woollen Felt, or Cotton Wool	·323	93	·290	71	° 107	56	·175	35·7	° 86
Sawdust (Mahogany) ..	·523	131	·409	100	123	86	·269	54·9	98
Fir-wood (transmission against grain)	·748	166	·519	127	137	119	·372	76·0	110
Coal Ashes (Coke pulverized)	1·29	225	·703	172	158	173	·541	110	129
Plaster, common	3·86	313	·978	239	188	299	·934	191	168
Stone	13·68	365	1·14	279	204	407	1·272	259	197
Marble, grey, fine	28·00	376	1·175	287	208	430	1·344	274	204

NOTE.—The ratio in this table gives the loss by the cased pipe, compared with with fir-wood casing, 2 inches thick, the loss per foot run is ·238, that by an

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is remarkable; thus a 4-inch pipe with 6 inches of casing has a surface temperature of 63°·58, or 3°·58 only above the atmosphere. The ratio in this table gives the comparative losses with cased and *naked* pipes (not covered with canvas as in Table 110); this is obtained by the same method as in (316), and thus we have

	R.	Ratio.	A.	Ratio.	Diff.	Circum. in Feet.	Per Foot run. Units.
For 2-inch pipe	(·7 ×	1·5) +	(·728	× 1·55)	× 152 ×	·5236	= 173·4
4 „	„		(·5745	× 1·55)	× 152 ×	1·0467	= 308·7
8 „	„		(·4978	× 1·55)	× 152 ×	2·094	= 576·5
12 „	„		(·4722	× 1·55)	× 152 ×	3·142	= 851·0

Table 111 shows that a 4-inch pipe cased in 2 inches of felt loses 34·7 units per foot run, and that the naked pipe would have lost 308·7 units, or 308·7 ÷ 34·7 = 8·9 times as much as

DIAMETER, heated to 212°, freely exposed to Air and Radiant Objects at 60°, Thin Canvas; showing the Effect of the varying Conducting Power the Loss of Heat.

PIPE 4 INCHES DIAMETER OUTSIDE.

2 Inches thick.				4 Inches thick.				6 Inches thick.			
Loss of Heat per Hour			Temp. of the External Surface.	Loss of Heat per Hour.			Temp. of the External Surface.	Loss of Heat per Hour.			Temp. of the External Surface.
Per Foot run.		Per Foot square.		Per Foot run.		Per Foot square.		Per Foot run.		Per Foot square.	
Units.	Ratio.	Units.		Units.	Ratio.	Units.		Units.	Ratio.	Units.	
			°				°				°
34·7	·1086	16·6	73	22·36	·070	7·124	65·88	18·10	·0566	4·323	63·58
54·8	·1725	26·2	80	35·86	·112	11·42	69·27	28·92	·0904	6·908	65·73
76·1	·2380	36·4	87	51·15	·160	16·29	73	41·06	·1283	9·81	68·04
124	·3875	59·5	101	85·7	·268	27·3	81	69	·2156	16·6	73·4
272	·8500	130	140	223	·697	71·0	109	201	·628	48	95
450	1·407	215	181	499	1·56	159	156	498	1·55	119	137
519	1·622	248	195	644	2·01	205	178	720	2·25	172	163

the loss by a pipe without casing, but covered with canvas like the others; thus uncased pipe being 1·0, &c. The uncased pipe loses 320 units per foot run.

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the cased pipe. This ratio is given for each diameter and thickness in the table, and may be used with sufficient correctness to find the loss by other sizes of pipes: thus with 2 inches of felt, the ratio for a 4-inch pipe is 8·9, and for 8-inch 9·71, a 6-inch one might therefore be taken at $(8\cdot9 + 9\cdot71) \div 2 = 9\cdot3$. Then by Table 90 the loss by a 6-inch naked pipe at 210° is 493 units per foot run; cased in 2 inches of felt, this would be reduced to $493 \div 9\cdot3 = 53$ units, &c. It will be observed that very thin casings of felt are effective in reducing the loss of heat; thus $\frac{1}{4}$ inch reduces the loss to less than half the amount with a naked pipe, &c. Great thicknesses are very effective, but they become unsightly from the great increase in bulk; one inch is a common thickness, and the loss is then about one-fifth of that with a naked pipe.

TABLE 111.—Of the Loss of HEAT by HORIZONTAL PIPES of different DIAMETERS cased in WOOLLEN FELT or COTTON WOOL, and covered with THIN CANVAS, &c.; the Pipes being heated to 212°, and freely exposed to External Air and Radiant Objects at 60°.

Thickness of the Felt or Cotton-wool Casing in Inches.	Diameter of the Pipe (outside, without Casing).															
	2 Inches Diameter.				4 Inches Diameter.				8 Inches Diameter.				12 Inches Diameter.			
	Loss of Heat per Hour.			Temp. of the External Surface.	Loss of Heat per Hour.			Temp. of the External Surface.	Loss of Heat per Hour.			Temp. of the External Surface.	Loss of Heat per Hour.			Temp. of the External Surface.
	Per Foot run.		Per Foot square.		Per Foot run.		Per Foot square.		Per Foot run.		Per Foot square.		Per Foot run.		Per Foot square.	
	Units.	Ratio.	Units.	°	Units.	Ratio.	Units.	°	Units.	Ratio.	Units.	°	Units.	Ratio.	Units.	°
0	173·4	1·00	331·0	212·	308·7	1·00	295·0	212·	576·5	1·00	275·0	212·	851·0	1·00	270·8	212·
¼	79·6	2·18	121·8	128·	142·6	2·16	121·2	133·	..	..	..	..	..	..	..	..
½	52·4	3 31	66·8	102·	93·	3·32	71·0	107	174·3	3·308	74·0	110·	236·4	3·60	69·50	108·
1	33·54	5·02	32·05	82·8	56·	5·50	35·7	86·	101·9	5·657	38·94	89·	148·2	5·74	40·46	90·
2	21·95	7·90	13·98	70·9	34·7	8·9	16·6	73·	59·37	9·71	18·91	75·	77·9	10·92	18·60	75·
4	15·47	11·20	5·91	64 8	22·4	13·8	7·12	65·9	37·07	15·55	8·85	67·35	47·87	17·78	9·146	67·63
6	..	..	..	..	18·1	17·0	4·32	63·58	27·28	21·14	5·21	64·35	36·00	23·64	5·734	64·8

NOTE.—The ratio in this table gives the loss of heat by the naked pipe, the amount lost by a cased one being 1·0; thus a 4-inch naked pipe loses 8·9 times as much heat per foot run as it would lose if cased in 2 inches of felt, or 13·8 times as much as with 4 inches of felt, &c.

CHAPTER XIV.

ON VENTILATION ETC.

(324.) "*Respiration, &c.*"—In the act of respiration oxygen derived from the atmospheric air combines with carbon and hydrogen given out from the lungs, the carbon being transformed into carbonic acid, and the hydrogen into water or vapour. It is shown in (57) that combustion by which heat is obtained in our furnaces is effected by similar combinations, so that respiration is a veritable act of combustion at a low temperature, and an amount of heat is produced proportional to the carbon and hydrogen consumed. According to M. Dumas, an ordinary man burns per hour a quantity of carbon and hydrogen equivalent to $\cdot 022$ lb. of carbon, and by (57) the heat developed will be $12906 \times \cdot 022 = 284$ units per hour. The heat thus developed serves to keep up the temperature of the body: if there were no means to prevent its accumulation the temperature would rise without limit, but such means are provided, and are so nicely adjusted that the body is maintained uniformly at 98° under all circumstances. The surplus heat is carried off 1st, by radiation to cold walls or other surrounding objects; 2nd, by contact of cold air; and 3rd, by perspiration, in which the fluids of the body are vapourized, and heat becomes latent (17). With moderate temperatures all three of these means are in operation, and only then is perfect comfort experienced: in extreme cold, perspiration is nearly suppressed, and all the heat is carried off by radiation and contact of cold air. When the temperature becomes moderate, *insensible* perspiration ensues, and the skin becomes moist; as the temperature rises, the amount vapourized becomes progressively greater, and when air and walls have attained the temperature of the body, or 98°, the loss of heat by radiation and contact of air is suppressed, and the whole of the animal heat has to be carried off by perspiration, which then becomes excessive and takes the form of sweating.

(325.) "*Ventilation.*"—

quired for ventilation, we must be guided by the several offices which that air has to fulfil. These are five in number: 1st, to supply the organs of respiration with the necessary amount of oxygen; 2nd, to carry off the vapour given out by the lungs and the body; 3rd, to dilute and carry off the natural exhalations from the skin; 4th, to carry off the animal heat not absorbed by radiation and vapourization; and 5th, to supply the lighting apparatus. The same air may possibly perform several of these offices simultaneously or consecutively.

(326.) "*Air required for Respiration.*"—An ordinary man in a state of repose makes about sixteen respirations per minute, each of 40 cubic inches according to Menzies; admitting therefore that air should not be respired a second time, we have $(16 \times 40 \times 60) \div 1728 = 22$ cubic feet of air thus vitiated per hour.

(327.) "*Air required to carry the Vapour.*"—We have just seen that the amount of vapour produced varies very much with the temperature, &c.; if the whole of the animal heat had to be thus carried off we should have $284 \div (1178 - 98) = \cdot 263$ lb. of vapour per hour; and if the whole surface of the body, or even a considerable portion of it were exposed, there would be no inconvenience, for by col. 3 of Table 79 a square foot at 98° evaporates about ·13 lb. per hour, and two square feet of exposed surface would suffice, but where the body is almost wholly covered, as is usual with us, there would be great inconvenience and even distress. M. Dumas gives the amount of vapour in ordinary cases and with moderate temperatures at ·0836 lb. per hour, or about one-third of the maximum quantity, and this, with air at 62° would require $\cdot 0836 \times (1178 - 62) = 93$ units of heat to vapourize it, so that $284 - 93 = 191$ units would have to be carried off by radiation and contact of cold air.

It is shown in (185) that to effect evaporation at low temperatures, a certain volume of air is necessary to dissolve and carry off the vapour. Say we have air at 62° and ·6 saturated (181), which would be a common condition: each cubic foot would therefore by col. 7 of Table 68 contain $\cdot 000881 \times \cdot 6 = \cdot 000528$ lb. of vapour; to saturate it would require $\cdot 000881 - 000528 = \cdot 000353$ lb. more, and the ·0836 lb. given out by respiration would saturate $\cdot 0836 \div \cdot 000353 = 237$ cubic feet,

and this is the minimum volume of air possible under these conditions.

(328.) "*Air required to carry off the Exhalations.*"—The amount of air necessary to dilute and carry off the natural exhalations from the skin, &c., varies very much with the character of the individual as to health and personal cleanliness: observation and experiment alone can determine the minimum necessary in each particular case. With a ventilation of 212 cubic feet per head per hour M. Péclet found a slight odour in a public school of 180 children of the age of seven or eight years. With the same ventilation there was a sensible odour in a cell of the Prison Mazas, which disappeared entirely with 350 cubic feet.

In hospitals, especially those for the treatment of infectious diseases, a very much larger volume of air is necessary; indeed, there are only two limitations to the quantity which may be advantageously used in such cases, namely, the difficulty of distributing a very large volume so as to avoid objectionable draughts, and the cost. With 530 cubic feet per head per hour there was a very perceptible odour in the wards of the Hospital Beaujon: with 880 cubic feet the odour disappeared, but as this rate was preceded by a very powerful ventilation by opening windows, &c., we may admit 1000 cubic feet as the minimum volume to prevent sensible odour in ordinary cases of sickness.

But there can be no doubt that when infected air has been so far diluted as to cause the odour to disappear, it may still contain the germs of disease in dangerous proportions, hence very large volumes are now admitted to be desirable, if not essential. Péclet seems to admit 2000 and Morin 2500 cubic feet per head per hour for ordinary cases, and the latter recommends as much as 5300 cubic feet in times of epidemic, &c.; which is perhaps excessive.

(329.) "*Air required to carry off the Animal Heat.*"—It is shown in (327) that 284 units of heat are generated per head per hour, and that 93 units being required to form the vapour emitted, there remain 191 units to be dissipated by radiation and contact of cold air. When a large room has

very few occupants, a very considerable proportion of this animal heat will be given out by radiation to the cold walls; but in a crowded room, the radiation of each individual is almost wholly suppressed by the surrounding crowd, and in that case the air alone has to carry off the surplus heat, and it will be heated to an extent proportional to the volume admitted.

If the whole of the 191 units has to be carried off by the air, and restricting the increase in temperature to 20°, which is perhaps as much as could be conveniently permitted in most cases, the least volume of air would be $191 \div (20 \times \cdot 238 \times \cdot 0761) = 527$ cubic feet per head per hour.

In the case (328) where only 212 cubic feet were allowed to the single occupant of a prison cell; if that air were heated 20° it would carry off only $\cdot 0761 \times 212 \times \cdot 238 \times 20 = 78$ units, leaving $191 - 78 = 113$ units to be absorbed by radiation.

These two cases may be taken as the extremes, and we thus find that to carry off the animal heat, the volume of air will vary from say 220 to 500 cubic feet per head per hour, depending on the more or less crowded state of the room.

(330.) "*Air for Lighting Apparatus.*"—According to Dr. Ure 1 lb. of coal-gas requires 14·58 lbs. of air, supposing that all the oxygen in that air is consumed; but a considerable portion will always escape unconsumed, and we may admit that a double volume (76) should be supplied, or 29 lbs. of air per pound of gas. Taking the specific gravity of coal-gas at ·42, that of air being 1·0, a cubic foot at 62° will weigh $\cdot 0761 \times \cdot 42 = \cdot 032$ lb., and will require $\cdot 032 \times 29 = 9 \cdot 28$ lbs., or $9 \cdot 28 \div \cdot 0761 = 12$ cubic feet of air. An ordinary gas-burner consumes about 5 cubic feet of gas per hour, and requires $5 \times 12 = 60$ cubic feet of air, and if we allow one gas-burner per head, we require 60 cubic feet per head for the lighting apparatus.

(331.) The combined and general result of this investigation is given by Table 112, from which we conclude that in a room *very* thinly occupied, the minimum volume of air with cleanly and healthy persons may be as low as say 250 cubic feet per head per hour; for prisons, workhouses, &c., 350; for crowded assembly rooms, chapels, &c., 500; for hospitals with ordinary maladies, 2000; and for fever hospitals, &c., 4000 cubic feet.

TABLE 112.—Of the CUBIC FEET of AIR required for the different purposes of VENTILATION.

Character of Occupants.	For Respiration.	For Vapour.	For Exhalation.	For Heat.	For Lights.
Room with single occupant, cleanly and healthy	22	237	250	220	60
Room with single occupant, healthy but not cleanly	22	237	350	220	60
Room with single occupant, cleanly but sick	22	237	1000	220	60
Crowded room, healthy and cleanly persons	22	237	250	500	60
Hospitals (ordinary cases)	22	237	2000	220	60
Hospitals for Fevers, &c.	22	237	4000	220	60

It should be observed that the same air serves simultaneously, or consecutively, all the five offices we have assigned to it in (325). Thus, the lowest volume, 250 cubic feet, suffices with cleanly and healthy persons, for respiration, 22; vapour, 237; the exhalations, 250; the animal heat, 220; and the lighting, 60 cubic feet.

The results of many observations on ventilated buildings by various observers are given in Table 113; Table 114 gives the volume of air necessary for different cases according to the authority of Péclet and Morin; those by the latter are in many cases excessive.

(332.) "*Method of effecting Ventilation.*"—We have now to consider the means by which the continuous renewal of the air in a room or building is to be effected. Let Fig. 93 be a room filled with heated air, and let the walls have the same temperature as the air; the temperature will not be the same throughout, but will vary slightly with the height, the top of the room being the hottest. If the walls are exposed to cooling influences, and become colder than the internal air, a down-current is established against the face of the wall, as shown by the arrows, Fig. 87, because the air in contact with the wall being cooled by it, becomes heavier than the air in the centre of the room, the former therefore descends, and the latter ascends, as shown

TABLE 113.—Of the VENTILATION of BUILDINGS; Cases in Practice.

Kind of Building.	Cubic Feet of Air per Head per Hour.			Observer or Authority.
	Max.	Min.	Mean.	
Hospital, Guy's, night	..	..	900	Mr. Rosser.
" " day	3960	2220	3090	"
" Lariboisière	3950	2000	2975	M. Grassi.
" Necker, Summer, entry	..	..	2084	MM. Combes, Leblanc, &c.
" " " exit	..	..	2472	"
" " Winter, entry	..	..	3108	"
" " " exit	..	..	3460	"
" Beaujon	..	..	530	Morin (sensible odour).
" "	..	..	880	" (odour disappeared).
" Vincennès (military)	4240	..	..	" (too much air; draughts).
" " "	..	1060	..	" (too little air).
" " "	..	..	2120	" (satisfactory).
Théâtre du Palais Royal	..	..	636	M. Darcet.
" Lyrique, per *seat*	1480	1150	1315	M. Morin.
" de la Gaité "	1327	925	1126	"
Prison cellulaire, Mazas	..	212	..	M. Péclet, &c. (sensible odour).
" " "	353	..	..	" (very slight odour).
" " de Provins	3072	908	1990	M. Gentilhomme.
" " de Tours	918	530	724	M. Sagey.
School, 120 children, Grenelle	..	83	..	Morin (bad odour, very sensible).
" 80 " "	..	..	120	" (" but slight).
" 76 " "	194	..	..	" (very little odour).
" 180 " " 7 to 10 years old	..	..	212	Péclet (very slight odour).
Salle des Séances de l'Institut	1050	996	1023	M. Cheronnet.
Chambre des Pairs	706	424	565	Gay-Lussac, Pouillet, &c.
Grand Amphithéâtre, Conservatoire des Arts, &c.	530	353	442	Morin.
Ancienne Chambre des Députés	..	..	635	Péclet (satisfactory, no odour).

TABLE 114.—Of the VOLUME of AIR for VENTILATION, according to different Authorities.

Place Ventilated.	Cubic Feet of Air per Head per Hour.			Authority.
	Max.	Min.	Mean.	
Hospitals	..	..	2120	Péclet.
Theatres, Assembly Rooms, &c.	..	..	530	"
Prisons	..	..	350	"
Ordinary rooms	390	212	300	"
Schools	..	..	212	"
Hospitals, ordinary maladies ..	..	..	2470	Morin.
" wounded, &c.	..	.	3530	"
" in times of epidemic	..	..	5300	"
Theatres	1760	1410	1585	"
Assembly Rooms, prolonged sittings	..	..	2120	"
Prisons	..	..	1760	"
Workshops, ordinary	..	..	2120	"
" insalubrious	..	..	3530	"
Barracks, during the day	..	..	1060	"
" " night ..	..	..	1760	"
Schools, infant	706	530	618	"
" adult	1410	1060	1235	"
Stables	7060	6350	6700	"

by the arrows; and by the same means horizontal currents are established along the floor and ceiling, as shown by the figure.

(333.) If openings be made in the floor and ceiling, as at A and B, Fig. 88, the column A B would immediately ascend, being lighter than the column of cold external air with the same height, and the velocity of the motion can be easily calculated. Say the room is 10 feet high, with internal air at 62° and external air at 32°. We find by Table 24 that a column of air 10 feet high at 62° is equal to a column at 32°, $10 \div 1 \cdot 061 = 9 \cdot 424$ feet high, we have therefore $10 - 9 \cdot 424 = \cdot 576$ foot head to produce motion, and as we have two openings to deal with, we may suppose it to be divided between them, each taking $\cdot 576 \div 2 = \cdot 288$ foot head, and as in the case (150) we have $\sqrt{\cdot 288 \times 8} = 4 \cdot 3$ feet per second, or with $\cdot 93$ for coefficient (153) $4 \cdot 3 \times \cdot 93 = 4$ feet per second, and from this the area necessary can be easily calculated.

(334.) But it is obvious, that although a given amount of air might thus be caused to pass through a room, we should not have efficient ventilation; the portions of the room indicated by C and D would not be ventilated at all. If, as in Fig. 89, the openings were made at opposite ends of the room, the ventilation would not be much improved, the central part C receiving no benefit from the current of fresh air passing through the room.

(335.) If, as in Fig. 90, we admit the fresh air by very numerous holes, equally distributed all over the floor, and allow it to escape by similar holes in the roof, we then have perfect ventilation; but the plan is very difficult to carry out. Thus in Fig. 91 we have holes in the floor J and ceiling K, as in Fig. 90; but admitting the air by one large opening, A, and allowing it to depart by a similar opening at B, which is usually a practical necessity, the air takes the shortest course, and the holes in the direct line get the most of it, and it is exceedingly difficult to obtain uniformity in the ventilation.

(336.) There is another and a practical difficulty in this mode of ventilation. It is essential that the numerous inlet openings should be of large area, otherwise the velocity of the air would be so great as to be a nuisance, especially for summer ventilation, when cold air is admitted; but if the velocity be very low, it is apt to be greatly disturbed, and the current even reversed by light winds in certain directions, so that it is essential that the velocity through the inlet and outlet openings should be considerable.

(337.) In Fig. 92 we have a room ventilated on a totally different principle. The heated air enters by one large opening at A, and mounts directly to the ceiling. The cooled air is drawn off by an opening B, leading to an air-chimney C, in which the air is maintained at all seasons at a high temperature, thus causing a rapid current, too strong to be seriously affected by winds. The heated air entering the room rises in a body to the roof, and is distributed in horizontal strata, having the same temperature at the same level all over the room, as in Fig. 93. This is a natural circumstance where there are no disturbing causes, and an important one for our purpose. Let Fig. 94 be a room like Fig. 93, but say that by some accidental

cause the air at one end of the room is more heated than at the other, the beds of air will in this case be at an angle with the horizon, as per figure. It will be seen that the *mean* temperature of the air at one end of the room is 10° higher than at the other, it will therefore weigh less, and motion will ensue in the direction of the arrows until uniform horizontal temperatures are obtained, as in Fig. 93. By the action of the ventilating shaft the air is regularly drawn down, and the number and position of the exit orifices (on the plan) has no influence on the ventilation. But if, as in summer, the air admitted at A is cooler than that in the room, this arrangement of the openings would be the worst possible; for, as in Fig. 95, the cold air would remain at the bottom, passing away along the floor to the opening B without ventilating the room at all. We require in this case an opening at the top of the room, or at least above the heads of the people, as at D. The cold air admitted at B will then spread horizontally all over the floor, and ascending regularly by the action of the chimney and by the heat imparted to it, will depart vitiated by the opening D. If instead of admitting the cold air at A, we let it in by an opening in the ceiling at E, the cold air would fall direct to the floor because of its superior weight, and would then be distributed and rise uniformly to the opening D as before; but the descent of the cold air would be felt as a draught by those in and near its course.

(338.) "*Summer and Winter Ventilation.*"—It will be seen from this that the arrangement of the orifices of access and exit of air must be varied with the seasons. In winter, when the air entering is previously heated, and is of higher temperature than the room, it may enter by one or more openings in any part of the room, and should be drawn off by one or more openings near the ground into the air-chimney. In summer the cool external air should be admitted by openings in the floor, and should pass into the chimney by openings near the top of the room, or at least above the heads of the inmates. For both cases we have supposed a good air-chimney to be used, and an active draught to be maintained, summer and winter, when ventilation is required.

(339.) With *natural ventilation*, as it is termed, as per Figs. 88–91, no change is necessary with the seasons. In all cases the air should enter by numerous openings in the floor, and depart by similar openings in the ceiling. No chimney is necessary, but the exit opening should be provided with a hood to prevent the action of the wind disturbing the draught. This mode *appears* by far the cheapest and simplest; but practically it is very uncertain and frequently very ineffective. It is almost the only plan used in our churches and chapels, and by universal experience is far from satisfactory, especially in summer, although supplemented by opening windows and other highly objectionable means. It is deeply to be regretted that the acknowledged difficulties of accomplishing effective ventilation have led to the whole question being virtually abandoned by architects and others designing many of our public and private buildings.

(340.) "*Mechanical Ventilation.*"—Ventilation may also be very effectively, and in many cases cheaply accomplished by a fan worked by manual or engine power. In factories, and other large establishments where an engine is used for other purposes, this is by far the best method of ventilating workshops, &c. The great objection to this method for general purposes, namely, that the ventilation continues only so long as the engine is at work, does not apply to such cases, the hours of labour terminating at the same time. In many cases, such as prisons and workhouses, manual labour may be substituted for engine power with advantage, the fan being worked by prisoners or paupers in short relays to prevent the labour being excessive. By a heavy weight wound up during the day and allowed by suitable machinery to descend slowly during the night, the ventilation may be kept up continuously. The same method may be adopted for the ventilation of churches and chapels; usually these buildings are occupied only three or four hours on Sundays, and the weight may be wound up during the preceding week.

(341.) There are two kinds of fan which may be used for ventilating purposes, one compressing, and the other exhausting the air operated on. The former is preferable in most cases; the latter having the objection that it causes draughts of cold

air to enter by all crevices in the floors, windows, and open doors, &c. It is evident from (338) that a compressing fan should deliver the air near the floor-line, the outlets from the building being near the floor-line in winter and near the ceiling in summer.

The general form and proportions of ventilating fans may be illustrated by Figs. 63 to 66, which give the construction and details of a 5-foot fan and the gearing for working it as adapted for the case of a chapel, workhouse, &c., where the ventilation has to be kept up by a falling weight.

(342.) Figs. 63 to 66 give all the necessary details of the fan and the gearing for working it. One of the principal things to be attended to is to make all the parts *very light*, and another is to avoid enlargements of the area in the air-passages, by which velocity is lost after having been got up elsewhere. The air-passages should either have the same area throughout, or, what is perhaps better still, a diminishing area, so that the velocity is gradually got up from point to point, but never suffered to fall again by an enlargement of channel: see (163). In Fig. 63, A is the fan, 5 feet diameter, having six blades of light sheet zinc or iron, which are curved to a radius of three-eighths of the diameter, or in our case 22½ inches, the describing circle or centre being one-third of the diameter, or 20 inches, as in the figure; we thus obtain air-passages of 7 inches at the smallest part between blade and blade, increasing to the circumference. The blades are supported by two sets of light deal arms, fixed to light cast-iron centre bosses, keyed on the wrought-iron shaft. The centre openings are half the diameter of the fan, or 30 inches, and the distance between the side cheeks is the same dimension. The area of the two central 30-inch openings is $9 \cdot 8$ feet; the area of the passages between the fan-blades at the narrowest part is $7 \times 30 \times 6 \div 144 = 8 \cdot 75$ feet; and the outlet opening is $3 \times 2 \cdot 5 = 7 \cdot 5$ square feet: the velocity is thus gradually got up. The case may be made of light deal or any other convenient material.

The gearing should be very light in pitch, and truly fitted in the teeth. The pinion B is made to slide on a feather let into the fan-shaft, so as to draw endwise out of gear with its wheel when the weight is being wound up, which is done by a

handle, C, on the second-motion shaft. This handle should be removed after the weight is wound up; and the machine is stopped till required by the break D, which is kept in contact with the unturned wheel F by the weight G. To start the machine, the break-lever is raised by its handle H, and is kept up by the spring holder J. The lever D is fitted with a block of hard wood, curved to fit the wheel F. The frames are secured to the floor by bolts in their feet, and to one another by a cross-stay at K. The barrel carries a light wire or hempen rope, which is led under a large guide-pulley, and thence over another at the top of the building, so as to raise the weight in a channel provided for it in the wall, being guided and kept in position there by long vertical guides of iron or hard wood.

(343.) The power which such a fan will require to drive it depends very much on the velocity of the current of air issuing from it; in fact, the power is directly as the square of the velocity omitting the consideration of the friction of the machine, so that velocities in the ratio 1, 2, 3, &c., require powers in the ratio 1, 4, 9, &c. There is a practical limit, however, to the extent to which the power may be reduced by reducing the speed, for at very low velocities the current is apt to be retarded or even overpowered and reversed by adverse winds.

It is shown in (394) that the velocity actually realized in practice is considerably less than that due to the actuating force, the ratio varying with the length of the passages, and the number of successive enlargements and contractions; on the large scale we may admit the ratio of the Prison Mazas, or $\cdot 423$ for the velocity, and $\cdot 423^2$ or $\cdot 179$ for the power required to drive the fan. Thus, say we required such a volume of air that the actual velocity through the passages must be 10 feet per second, and the theoretical power for that velocity was 1-horse; then the speed of the fan must be calculated for $10 \div \cdot 423 = 24$ feet per second, and the power of the engine would be $1 \div \cdot 423^2 = 5 \cdot 6$ horses. But for most cases the passages being much shorter than in the Prison Mazas, we may admit the ratio of the real to the theoretical velocity to be half, or $\cdot 5$, and the power $\cdot 5^2 = \cdot 25$: admitting

this, and assuming 5 feet per second as a good *real* velocity, we must calculate for $5 \div \cdot 5 = 10$ feet per second in determining the revolutions of the fan and the power required to drive it, the real velocity of discharge, however, being only 5 feet per second. Then with a velocity of 5 feet per second, or 300 feet per minute, the fan, Fig. 63, having an exit opening of 7·5 square feet, would discharge $7 \cdot 5 \times 300 = 2250$ cubic feet per minute, or by Table 24, $\cdot 0761 \times 2250 = 171$ lbs. of air. The head due to 10 feet velocity by the laws of falling bodies, is $(10 \div 8)^2 = 1 \cdot 56$ foot, and the mechanical work to be done is to raise 171 lbs. of air 1·56 foot high per minute, or $171 \times 1 \cdot 56 = 267$ foot-pounds. Table 115, which gives the proportions of ventilating fans, has been calculated in this way. The centre of effort of a fan is about seven-eighths of the extreme diameter, or in our case $5 \times 7 \div 8 = 4 \cdot 35$ feet, the circumference of which is $4 \cdot 35 \times 3 \cdot 14 = 13 \cdot 7$ feet, and for 10 feet per second, or 600 feet per minute, we shall require $600 \div 13 \cdot 7 = 44$ revolutions per minute.

TABLE 115.—Of the PROPORTIONS of VENTILATING FANS.

Diameter of Fan in Feet.	Size of Exit Opening in Feet.			Velocity 5 Feet per Second			Velocity 10 Feet per Second.		
	Width.	Length.	Area.	Revolutions per Minute.	Cubic Feet of Air per Minute.	Foot-lbs. per Minute.	Revolutions per Minute.	Cubic Feet of Air per Minute.	Foot lbs. per Minute.
3	1·5	1·8	2·7	73	810	96	146	1620	384
4	2·0	2·4	4·8	54	1440	171	108	2880	684
5	2·5	3·0	7·5	44	2250	267	88	4500	1068
6	3·0	3·6	10·8	37	3240	385	74	6480	1540
7	3·5	4·2	14·7	32	4410	524	64	8820	2096
8	4·0	4·8	19·2	28	5760	685	56	11520	2740
9	4·5	5·4	24·3	25	7290	867	50	14580	3468
10	5·0	6·0	30·0	22	9000	1070	44	18000	4280
12	6·0	7·2	43·2	18	12960	1540	36	25920	6160
15	7·5	9·0	67·5	15	20250	2403	30	40500	9610
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)

(344.) In cases where the economy of power is not a matter of primary importance, a greater velocity may be permitted; thus with a double velocity, or 10 feet per second, of course a double volume of air will be discharged, and the power

quadrupled, becoming $267 \times 4 = 1068$ foot-pounds, as in col. 10.

With a 5-foot fan discharging air with velocities of 5, 10, 15, and 20 feet per second, we require 44, 88, 132, and 176 revolutions per minute, and power equal to 267, 1068, 2403, and 4272 foot-pounds respectively.

An ordinary able-bodied man can easily exert 3000 foot-pounds per minute, working at a winch for six hours per day, but allowing only half with such men as prisoners, &c., or 1500 foot-pounds, cols. 7 and 10 show that a single man could work a 12-foot fan at 5 feet velocity, or a 6-foot fan at 10 feet per second.

(345.) "*Examples.*"—The application of the rules and principles we have arrived at in this and preceding chapters will be best understood by examples. We will take three cases; the first being a school, in which the air is heated and ventilation effected by stoves; the second, a chapel, heated by ordinary hot-water pipes, and ventilated by a compressing fan driven by a weight wound up by manual labour during the preceding week; and the third, a hospital, heated by hot-air pipes and ventilated by a draught-chimney with a fire maintained at its base in all seasons.

(346.) It is important to observe that the maximum power of the heating apparatus must be fixed for the greatest ordinary difference of external and internal temperature: this may be taken at 30°, or say internal air at 60°, and external air at 30°. But the *mean* temperature of the six cold months of the year during which the heating apparatus is required, or October to March inclusive, is by Table 33 about 42°: the difference is then $60 - 42 = 18°$ only, and the mean consumption of fuel will be governed by that difference.

(347.) "*Schools, &c.*"—Figs. 96–98 show a school for 100 boys, on the plan proposed by Péclet, heated and ventilated by a stove; this is constructed with a double case, the inner one containing the fire, and the smoke passes by a long pipe to a chimney, N, at the end of the room; the air to be heated passes between the two cases, being drawn from the room itself when the fires are first lighted by closing the

damper B and opening the door at A; but when *ventilation* is required during school hours, the damper B is opened wide, and A is closed, the air being drawn through a grating at C by the channel D from the exterior. The air between the cases being highly heated, we have a column the height of the stove dilated to say one-third the volume of the air in the room, and a powerful current is created, by which air is *forced* into the room, passing out by openings at the top of the case. The air *in the room* is further heated by the flue-pipes, and in both cases it will rise to the ceiling by reason of its levity, and would pass away if openings were made at the top of the room, and its heat would be wasted; but openings for ventilation are made about 18 inches or 2 feet from the ground at E F, which can be opened and closed at pleasure. The air is therefore distributed all over the room in horizontal layers, having the same temperature throughout (337), and is regularly drawn down, becoming cooler as it descends by giving out its heat to the walls. The walls are also further heated by *radiation* from the stove and pipes.

(348.) For summer ventilation, when of course no fire is needed in the stove, a small special fire is maintained at the base of the chimney N by a furnace H, the openings E and F are closed for reasons given in (337), and a large register opening, G, is opened by the cords J J; this opening may be as near the top of the room as convenient; it must be above the heads of the inmates, as we have seen in (337).

The only objection to placing it quite at the top of the room, is that the effective height of the chimney and the power of the draught are reduced thereby; the *effective* height, in fact, is the distance G L, which of course is reduced by placing G at a higher level.

(349.) In the cold season, the fire must be lighted an hour or two before school hours, in order to warm the walls and air in the room, which indeed is the principal object of the heating apparatus; for, as we shall see (356), each individual emits heat enough to warm the air required for ventilation, and the fire may die out an hour or so before the time of dismission, the walls, &c., containing a volume of heat that is not quickly dis-

sipated. When first lighted, *all* the exit openings, E F G, as also the furnace door at H and its register K, are closed, the stove heating the air in the room only, and this enters by the door A, the damper B being closed. When the pupils have assembled, B E and F are opened, and A is closed, &c., &c. The fire is regulated by a damper at M.

In summer the fire need not be lighted in H before the hour of assembling, K is opened only sufficiently to maintain the fire in a fair state of combustion, A E and F are closed, and B and G opened.

We may calculate the proportions of the entire apparatus by the rules already given.

(350.) "*Heat dissipated by the Walls, &c.*"—The building exposes an area of 1340 square feet of 14-inch walls, and 210 square feet of windows; with an internal temperature of 60°, and external 30°, the walls by Table 102 will lose $1340 \times \cdot 159 \times 30 = 6400$ units per hour, and the loss by the windows being by (307) 12·3 units per square foot, will be $12 \cdot 3 \times 210 = 2583$ units per hour, giving a total of 8983 units per hour as the loss by the building after it has attained the standard temperature.

But a large amount of heat must be absorbed by the walls before that standard temperature can be attained, and the proportions of the heating apparatus must be fixed with special reference to the preliminary heating of the building. By (293) and Fig. 80, with 14-inch walls, internal air at 60°, and external 20°, the mean temperature of the mass of the walls is $(48 \cdot 02 + 34 \cdot 2) \div 2 = 41° \cdot 11$ say 41°, they have therefore to be heated in the morning from 30° to 41°, or 11°, and as they contain about 1570 cubic feet of brickwork, weighing by Table 37, 115 lbs. per cubic foot, and the specific heat of brickwork (burnt clay) being ·185 by Table 1, they will require $1570 \times 11 \times \cdot 185 = 367420$ units of heat to raise their temperature from 30° to 41°. In our case they will receive it from two sources: from the heated air in the room, and by direct radiation from the stove and stove-pipes.

(351.) The amount that *can* be received by contact of heated air, is at first, when the walls are cold, or at 30°, by Table 97, $= \cdot 435 \times 30 = 13$ units per square foot per hour, but at the

end of the operation, when the walls are heated to the standard internal temperature of 48°, we have $\cdot 435 \times (60 - 48) = 5\cdot 2$ units; the mean is $(13 + 5\cdot 2) \div 2 = 9\cdot 1$ units per square foot per hour. A highly heated stove-pipe under ordinary conditions (248) gives out 74 per cent. by radiation to the walls, and 26 per cent. to the air in contact with it. Now, the maximum heat which the walls at 30° can receive from air at 60°, is $13 \times 1340 = 17420$ units per hour, and this being 26 per cent. of the total heat given out by the pipe, the latter must be $17420 \times 100 \div 26 = 67000$ units per hour; so that in this case $67000 - 17420 = 49580$ units are given out to the walls by direct radiation, and 17,420 units to the air, by which it is afterwards given to the walls. With the ordinary high temperature of the stove-pipe, the amount of heat given out will not vary sensibly with the slight variation of the internal surface of the wall from 30° to 41°, and they will receive throughout the heating process 67,000 units per hour.

But while this is going on, heat will be dissipated by the external surface of the wall; this is nothing at first, the wall having the same temperature as the external air, but when the standard temperature is attained, it becomes 8983 units per hour, as we have seen (350), the mean is $(8983 + 0) \div 2 = 4492$ units, hence the walls retain $67000 - 4492 = 62508$ units per hour, and to raise them to the standard temperature we shall require $367420 \div 62508 = 5\cdot 8$ hours.

(352.) It will be instructive to observe here, that the time required to heat the *walls* of a building is much less with stoves whose temperature is very high, than with hot-water or steam pipes whose temperature is comparatively low; and there is a special reason for this. At low temperatures pipes give out their heat by radiation to walls, and by contact of air in nearly equal proportions, as is shown by (248) and Table 88, with hot-water pipes freely exposed in the room, the walls receiving, as we have seen, 13 units per square foot from the air, would receive at the same time about 13 units by radiation, or 26 units altogether at the commencement of the heating process and $5\cdot 2 \times 2 = 10\cdot 4$ units at the end of it; the mean is $(26 + 10\cdot 4) \div 2 = 18\cdot 2$ units per square foot, and the walls will receive $18\cdot 2 \times 1340 =$

24388 units per hour. The building will dissipate 4492 units per hour on external objects as before, hence the walls retain $24388 - 4492 = 19896$ units per hour, and to attain the standard temperature we should require $367420 \div 19896 = 18 \cdot 5$ hours with steam or hot-water pipes heated to 210° and freely exposed in the room to be heated.

(353.) If the hot-water pipes were *enclosed* in chambers, or channels under the floor in the common way, the time would be still greater, because in that case the radiation from them would be suppressed, or nearly so, and the walls would then receive heat only by contact of heated air, the maximum (351) being 13 units at first, which is reduced to $5 \cdot 2$ units at last. The mean being $(13 + 5 \cdot 2) \div 2 = 9 \cdot 1$ units, we have $9 \cdot 1 \times 1340 =$ 12060 units per hour, and as they still dissipate 4492, they retain $12060 - 4492 = 7568$ units per hour, and the time required to attain the standard temperature would be $367420 \div 7568 = 48 \cdot 5$ hours; nor would the time be shortened by increasing the power of the heating apparatus so long as the internal air is not suffered to exceed 60°: see (351), (359).

(354.) With a stove and flue-pipe having the proportions given in (249), each pound of coal gives out 3090 units by the body of the stove, and 8547 units by the flue-pipe; the total is $3090 + 8547 = 11637$ units, we shall therefore require $67000 \div 11637 = 6$ lbs. of coal per hour. By (249) the stove-pipe must have $6 \cdot 5 \times 6 = 39$ square feet of surface at the least: if the pipe alone had to do all the work we should have required by (247), $67000 \div 1317 = 51$ square feet, but having allowed that the body of the stove does one-fourth of the work, the surface of the pipe itself is reduced to $51 \times 3 \div 4 = 39$ square feet, as we have seen.

This consumption of fuel, 6 lbs. per hour, is the maximum quantity for the coldest weather, with 30° difference of internal and external temperature; by (346) the *mean* consumption throughout the six months of cold weather will be $6 \times 18 \div 30 = 3 \cdot 6$ lbs. per hour, which for $5 \cdot 8$ hours per day (351) and 26 days per month, is equal to $3 \cdot 6 \times 5 \cdot 8 \times 26 \times 6 \div 2240 = 1 \cdot 45$ ton of coal for that season (358).

(355.) "*Time to cool down the Building.*"—The time to cool the

building down to the external temperature is much greater than that required to heat it. The loss at first is 8983 units (350) gradually reducing to nothing, the mean being $(8983+0) \div 2 = 4492$ units per hour, and the building will cool down in $367420 \div 4492 = 82$ hours. In most cases about sixteen hours only elapse before the fire is relighted, so that what really happens is that the walls retain a large portion of their heat continually, and require so much less time to raise them to the standard temperature.

(356.) "*Ventilation.*"—The experiments of Péclet and Morin in Table 113 show that 212 cubic feet per head per hour is the minimum volume of air for ordinary cases. Allowing 220 cubic feet or $220 \times \cdot 076 = 16 \cdot 72$ lbs., we require to heat it 30°, $16 \cdot 72 \times 30 \times \cdot 238 \times 100 = 11938$ units per hour, which added to that dissipated by the building makes a total of $11938 + 8983 = 20921$ units per hour. But by (327) $191 \times 100 = 19100$ units will be emitted by the occupants, or nearly the amount required for ventilation and the building, agreeing with experience that in a crowded building artificial heat is not necessary except to warm the building beforehand.

(357.) "*Area of Inlet and Outlet Openings, &c.*"—The area of the openings must be calculated for summer rather than for winter ventilation, because the heat required to produce the former has to be obtained by a special fire maintained for no other purpose; the less the air can be heated, the less costly is the operation, but with low temperatures large apertures are necessary.

Let us assume in our case that the air in the chimney N is heated 50° above the temperature of the room and external air, which may both have a summer temperature of say 70°, the air in the chimney will therefore be at 120°, and the *effective* height G L being about 12 feet, this will be equal by Table 24 to $12 \times 1 \cdot 082 \div 1 \cdot 184 = 10 \cdot 97$ feet of exterior air, and giving $12 - 10 \cdot 97 = 1 \cdot 03$ foot for generating velocity (150). As this has to be given at two places (inlet and outlet), we must divide the available head into two equal portions, allowing $1 \cdot 03 \div 2 = \cdot 515$ foot for each, and the theoretical velocity will be $\sqrt{\cdot 515} \times 8 = 5 \cdot 74$ feet per second, but with an orifice in a

thin plate we have seen (153) that the real discharge is $\cdot 65$ of the theoretical, and we have $5 \cdot 74 \times \cdot 65 = 3 \cdot 73$ feet per second. and as we have $220 \times 100 \div 3600 = 6 \cdot 11$ cubic feet of air per second, the openings for summer ventilation must have an area of $6 \cdot 11 \div 3 \cdot 73 = 1 \cdot 64$ square foot. The circular inlet tunnel D must therefore be 18 inches diameter, and the register G, which exposes rather less than half its total area as available openings (say $\cdot 4$), must have a *total* area of $1 \cdot 64 \div \cdot 4 = 4 \cdot 1$ square feet, or say 2 feet 3 inches diameter.

(358.) The quantity of fuel required in the furnace at H can be easily calculated, it being the amount necessary to heat $16 \cdot 72 \times 100 = 1672$ lbs. of air 50°, or $1672 \times 50 \times \cdot 238 = 19697$ units, and allowing 12,000 units per pound of coal as per Morin's experiments (383), we shall require $19697 \div 12000 = 1 \cdot 6$ lb. of coal per hour, or say for 5 hours per day, 26 days per month, and 6 months of summer ventilation, $1 \cdot 6 \times 5 \times 26 \times 6 \div 2240 = \cdot 55$ ton. This is a small consumption, but Péclet found by observation that $2 \cdot 2$ lbs. of coal per hour was sufficient for ventilation with 200 children, allowing 212 cubic feet of air per head per hour, the air-chimney being 33 feet high, and 9 square feet in area.

In order to obtain sufficient inlet area at the stove for summer ventilation, the damper M should be closed, and the door A, as well as the furnace door, may be thrown open or removed for the season.

We found in (354) the consumption of fuel for winter heating, &c., to be $1 \cdot 45$ ton, and that for summer ventilation being $\cdot 55$ ton, we have a total of 2 tons of coal per year.

These particulars of apparatus for a school of 100 children may be easily applied by proportion for other sizes; but some care is necessary here, for it should be observed that the sizes of the building are not simply proportional to the number of children, for supposing the height of the room to be the same, buildings in the ratio 1, 2, 3, would hold scholars in the ratio 1, 4, 9, &c., and while the consumption of fuel for ventilation would be simply proportional to the number of children, that required for heating the school-room would be proportional to the size of the building.

Péclet found that with schools for 50, 100, 150, 200, 250, and 300 children, the consumption of coals for the coldest weather was 4·4, 6·6, 8·8, 11·0, 13·2, and 15·4 lbs. of coal per hour respectively, which is at the rate per 100 children, of 8·8, 6·6, 5·8, 5·5, 5·3, and 5·1 lbs. of coal.

(359.) "*Chapels, &c.*"—The case of a church or chapel used only one day per week, and that for a few hours only, is peculiar: the whole building exposed for six days to cooling influences will have taken pretty nearly the temperature of the external atmosphere, and will require a large amount of heat to bring the *walls*, &c., up to the standard temperature. To receive this heat, a considerable time will be required, whatever the power of the heating apparatus (351), and this question of time must be considered, in order to understand the whole matter. Moreover, the condition of the occupants is peculiar; for the most part they retain their outdoor attire, or a considerable portion of it, and do not require so great a degree of warmth as is necessary in ordinary dwellings; and further, when the building is crowded, the natural animal heat is alone sufficient to keep up a comfortable degree of warmth.

The case of a church of large dimensions is investigated in (398); we will now consider that of a chapel to hold 400 persons, having the sizes and form shown by Figs. 101 to 108.

(360.) "*Heating Apparatus.*"—The heating apparatus has to do two things: it has first to heat the walls of the building before occupation; and to warm the air before entering the building to supply 400 people during the hours of public worship. Assuming an extreme case, we may take the external temperature of a winter day at 30°, and the final temperature of the internal air after receiving the heat of the audience to be about 60°; the conditions are therefore similar to Fig. 80, the building being exposed on all sides to cooling influences.

(361.) The walls being 30° at first, and the internal air 60°, taking the value of A from Table 97 at ·4133, will absorb $\cdot 4133 \times 30 = 12 \cdot 4$ units per square foot. The area of the walls is about 3258 feet inside and 3368 feet outside (deducting the windows and doors), taking the mean, which is near enough for our purpose, we have 3313 square feet, which will absorb

$3313 \times 12 \cdot 4 = 41081$ units per hour. Then the windows, say 20 in number, each $6 \times 4 = 480$ square feet, by (307) will lose $12 \cdot 3$ units per square foot, or $480 \times 12 \cdot 3 = 5904$ units per hour, making together $41081 + 5904 = 46985$ or say 47000 units.

(362.) By Table 90 an *enclosed* pipe 3 inches diameter, heated to 200° with air at 60°, gives 184 units per foot run, but with air at 30°, as in our case, this will be increased to $\frac{184 \times (200 - 30^\circ)}{200 - 60^\circ}$ $= 223$ units per foot, and we shall require $47000 \div 223 = 210$ feet of 3-inch pipe, which may be arranged in two lines as in Fig. 104. These pipes are carried by rollers, like Fig. 53, which are supported on cross-beams, H, Fig. 107, built into the brick side-walls of the channels in which the pipes are enclosed.

The boiler may be of the common horse-shoe form, and should be placed below the level of the chapel floor so as to obtain a simple circulation (267); where this is quite impracticable it is *possible* to work a coil of pipes by a boiler placed above it, as in (273); in that case the feed-cistern must be placed in the roof at A, or perhaps at B in Fig. 106; but this plan should never be adopted if it can possibly be avoided. For the maximum of 47,000 units we should require about $47000 \div 4000 = 12$ lbs. of coal per hour (367), and a square foot of fire-grate (127). With a very short and small boiler, such as is commonly used for such purposes, we should not obtain usefully much more than the radiant heat (96) in the fuel, and much of that will be lost, so that we have taken in the above the low economic value of 4000 units per pound of coal (112).

(363.) "*Time required to heat the Apparatus itself, &c.*"—The water-pipes, boiler, &c., will never be cooled down to 30°, the external temperature. We have seen in (43), that at a depth of about 20 feet we come to a stratum having constantly the mean temperature of the year at that place, and when the earth is covered by a building and protected from the cooling action of the atmosphere, the ground will have a temperature varying very little from the same temperature (290). By Table 34 the mean temperature of the year at Greenwich is $49^\circ \cdot 2$, and we may take the minimum temperature of the pipes, &c., at 50°

The boiler and pipes contain about 3700 lbs. of cast iron and 1100 lbs. of water; to raise the water 160°, or from 50° to 210°, will require $1100 \times 160 = 176000$ units. Cast iron having by Table 1 a specific heat of ·13, we shall require $3700 \times 160 \times \cdot 13 = 76960$ units for the boiler and pipes. But while heat is being received from the fuel, a considerable amount will be given out to the air; at first this will be 0 and at last 47,000 units, the mean is therefore $(47000 + 0) \div 2 = 23500$ units per hour; while 47,000 units are received, 23,500 units are parted with, and 23,500 units are retained, and to raise the whole apparatus to its maximum temperature of 210° we shall require $(176000 + 76960) \div 23500 = 10 \cdot 8$ hours; but during that time it will give out to the air and the walls, &c., half its maximum amount of heat. This time might be reduced probably to nearly one-half by forcing the fire.

(364.) "*Time required for heating the Walls, &c.*"—By Fig. 80 it will be seen that when the standard temperature is attained, the internal surface of the wall will be 48° and the external surface 34°; the mean temperature of the wall will therefore be $(48 + 34) \div 2 = 41°$, or 11° higher than the external air. The building contains about 451,000 lbs. of brickwork, the specific heat of which being ·185 by Table 1, we shall require $451000 \times 11 \times \cdot 185 = 917780$ units to bring them to the standard temperature. When that temperature is attained, the difference between the air and the walls will be $60 - 48 = 12°$, and the walls will receive $\cdot 4133 \times 12 = 4 \cdot 96$ units per square foot per hour, and the same amount will be dissipated by the external surface. But at first the walls will receive, as we have seen (361), 12·4 units and the external surface will dissipate 0. The *mean* heat received is therefore $(12 \cdot 4 + 4 \cdot 96) \div 2 = 8 \cdot 68$ units, and the mean heat dissipated is $(4 \cdot 96 + 0) \div 2 = 2 \cdot 5$ units, so that $8 \cdot 68 - 2 \cdot 5 = 6 \cdot 18$ units are retained by the wall, or $3313 \times 6 \cdot 18 = 20474$ units per hour by the whole surface of the walls, and to obtain the standard temperature we shall require $917780 \div 20474 = 44 \cdot 8$ hours, the heating apparatus giving its full maximum effect all the time. During the time that the apparatus itself was being heated, we found that it gave out to the building only half its maximum effect, so that the

10·8 hours at half power is equivalent nearly to 5·4 hours of full power, and 5·4 hours go for nothing, and the time required from lighting the fire is $44 \cdot 8 + 5 \cdot 4 = 50 \cdot 2$ hours.

(365.) "*Time required to cool the Walls, &c., to the External Temperature.*"—When the fire is put out, the building will gradually cool down to the temperature of the external air. At first it will lose 4·96 units per square foot per hour, and at last 0, the mean loss is $(4 \cdot 96 + 0) \div 2 = 2 \cdot 5$ units, and for the whole surface of walls $3313 \times 2 \cdot 5 = 8282$ units per hour. To this has to be added the heat lost by the windows, which at first is 12·3 units, and at last 0, the mean being $(12 \cdot 3 + 0) \div 2 = 6 \cdot 2$ units per foot, or $480 \times 6 \cdot 2 = 2976$ units per hour. We found by (363) that the water in the apparatus required 176,000 units, and the iron in the pipes, &c., 76,960 units to heat it, and these will of course give out the same amount in cooling down. The walls will give out 917,780 units as in (364), so that the time to cool down is $\frac{176000 + 76960 + 917780}{8282 + 2976}$ $= 104$ hours, or 4·3 days.

(366.) "*Heat required to keep up the Temperature of the Building.*"—When the building is once raised to its standard temperature, a much smaller amount of heat per hour will suffice to maintain it. The walls receive, transmit, and dissipate on external objects 4·96 units per foot, as we have seen (364), and the total loss by the walls is $3313 \times 4 \cdot 96 = 16432$ units per hour. The windows lose $480 \times 12 \cdot 3 = 5904$ units, making from both sources $16432 + 5904 = 22336$, requiring $22336 \div 4000 = 5 \cdot 6$ lbs. of coal instead of 12 lbs., the amount used in getting up the temperature (367). During the hours of worship, a large amount of heat is carried off by the air required by the congregation, which is forced through the building by a fan and passes away highly heated into the atmosphere, but this loss of heat is more than supplied by the animal heat emitted, which being by (327) 191 units per head, we shall have from this source $191 \times 400 = 76400$ units, or nearly $3\frac{1}{2}$ times the amount required to keep up the temperature of the walls, &c., and $76400 \div 47000 = 1 \cdot 6$ time the maximum power of the heating apparatus.

(367.) The whole week of 168 hours may be divided into three portions: 50 hours being spent in getting up the temperature, 14 hours in maintaining it during public worship, and 104 hours in cooling down again. The weekly consumption of coals would be $(50 \times 12) + (14 \times 5 \cdot 6) = 678$ lbs., or 6 cwt.; if the temperature were maintained throughout the week by continuous slow firing, the consumption would be $168 \times 5 \cdot 6 = 940$ lbs., or $8 \cdot 4$ cwt. With regular and slow firing the economy of fuel would be considerable, and the actual consumption would probably be not more than $7\frac{1}{2}$ cwt., or $1\frac{1}{2}$ cwt. more than by intermittent firing: the chapel would always be ready for week-night and occasional services, and the conveniences of this mode of heating are so great that it should become general.

The consumption of fuel we have calculated above, and in (362) and (366) is for the coldest weather, with 30° difference of internal and external temperature. But by (346) the *mean* consumption throughout the cold season will be less than the maximum, in the ratio of 18 to 30: thus the mean weekly consumption would be about $6 \times 18 \div 30 = 3 \cdot 6$ cwt., or for the whole season of 6 months, $= 3 \cdot 6 \times 26 \div 20 = 4 \cdot 68$ tons of coal.

(368.) "*Ventilation.*"—Allowing for the maximum case of a crowded congregation 500 cubic feet per head per hour (331), we require $500 \times 400 = 200000$ cubic feet, or by Table 24, $\cdot 0761 \times 200000 = 15220$ lbs. of air, whose temperature would be increased by the heating apparatus $47000 \div (15220 \times \cdot 238) = 13°$, or from the external temperature to $30 + 13 = 43°$, which, although low, would be tolerable to a warmly-clad and crowded audience. This air would be further heated by the animal heat; receiving $191 \times 400 = 76400$ units, it would be heated $76400 \div (15220 \times \cdot 238) = 21°$, or to $43 + 21 = 64°$.

But, say that the chapel was half filled only; in that case 43° would be too low a temperature for the air at entry; this, however, would be easily obviated by reducing the supply of air in proportion to the numbers of the audience. We shall now require only half the former quantity, which, receiving the whole of the 47,000 units given out by the pipes, would be heated 26° instead of 13°, or to $30 + 26 = 56°$, which would be

a comfortable temperature. With so thin a congregation, much of the animal heat, perhaps one-third, would be given out to the walls by radiation (329), and only $191 \times 2 \div 3 = 127$ units per head to the air; we have then $127 \times 200 = 25400$ units per hour given to 7610 lbs. of air, whose temperature would be raised $25400 \div (7610 \times \cdot 238) = 14°$, or to $56 + 14 = 70°$.

(369.) The maximum quantity of air is $200000 \div 60 = 3333$ cubic feet per minute, which with a velocity of 5 feet per second would require, by col. 6 of Table 115, a fan 6 feet diameter, and a driving power of 385 foot-pounds per minute, and for say four hours, the duration of the services, this is equal to $385 \times 60 \times 4 = 92400$ foot-pounds, and if the weight descends 30 feet we require $92400 \div 30 = 3080$ lbs., or 27 cwt. An ordinary man can easily raise 3000 foot-pounds per minute by a winch, and could therefore raise the weight in $92400 \div 3000 = 31$ minutes, and of course this could be done at any time during the week.

With the reduced volume necessary for a thin congregation, the velocity of the fan would require to be reduced proportionately. Thus, with half the volume, as in (368), we require only $37 \div 2 = 18 \cdot 5$ revolutions, and the weight would be reduced to one-fourth, or $27 \div 4 = 7$ cwt. This would be easily effected by making the weight in divisions, and raising more or less, as may be found necessary by experience. The velocity might also be reduced by regulating the weights G on the break-lever D, but this would involve a waste of power.

With so slow a speed as 37 revolutions of the fan, there should be no objectionable noise with well-fitted wheels; but to avoid all risk it will be well that the wheel A be provided with wooden teeth. The strength and sizes of all the wheels must vary with the respective strains upon them; the pitch should be fine, and the requisite strength obtained by width on the face.

(370.) "*Air-passages, &c.*"—The air from the fan descends below the floor of the boiler-house, and, branching right and left, enters the channels in which the hot-water pipes are fixed, A, B, C, Fig. 104, from which it is conveyed by branch channels D, E, &c., into four other channels, G, H, J, K, which deliver the air by apertures under the seats in each pew, as at

L, M, N, O, &c. The arrangement of these openings is a matter of importance to avoid objectionable draughts. Figs. 101 and 102 illustrate two modes of doing it, which may be varied to suit local circumstances, &c. The plan 102 is much the best.

To supply air to the galleries, channels are taken from G and K, as at P, P, P, &c., and are continued by other channels, R, R, &c., in the walls, and thence into the pews by branches S, S, S, &c. To prevent the accumulation of heated air under the galleries, short channels are made at intervals, as at T, T, T, &c., covered with ornamental open gratings. The heated air passes away out of the building by channels W, W, &c., which deliver into the roof, where it finally escapes by the cowl V. In winter the heated air escapes by openings at Y. The reason for placing it at so low a level is that the heat may be retained as long as possible, for the purpose of heating the building. In summer the air may escape by openings near the ceiling at X. These openings, Y and X, must be closed by sliding registers, which should be connected together, so that when one is open the other is closed, &c. For summer ventilation the openings might be made through the ceiling at Z Z, which must be closed in winter. For the inlet of air to the fan a permanent provision should be made by a louvréd window U, having a *free* area of about 12 square feet, and well-fitted shutters should be provided to close it completely while the walls are being heated (364).

(371.) The area of the channels should be proportional to the volume of air passing through them. With a velocity of 5 feet per second, or 300 feet per minute, and 500 cubic feet of air per head, each of the main channels as they leave the fan should have an area of $(500 \times 400) \div (2 \times 60 \times 300) = 5 \cdot 5$ square feet, diminishing regularly towards the remote end as the volume of air is reduced by the side channels D, E, &c.; the best method being to maintain a uniform width and reduce the depth. A section of these channels is given in Fig. 107. The pipes are carried by rollers on cross-timbers H, built at intervals across the channels, and they should coincide with the branch channels D so that the contraction of area from both

may occur at one place, and then to obviate it and preserve the area, the bottom should dip in a curve, as in Fig. 108.

The hot-water pipes are enclosed completely in a wooden casing, the air is admitted by narrow slits under the whole length of each pipe, the total area of which must be equal to the area of the two main channels, or 11 square feet = 1584 square inches, and the length being 214 feet, or 2568 inches, and allowing ·8 for the coefficient of contraction (153), the width must be $1584 \div (2568 \times \cdot 8) = \cdot 77$, or say $\frac{3}{4}$ inch.

(372.) In fixing the area of the branch channels, we may allow $1584 \div 400 = 4$ square inches per head. The galleries may contain 90 people, and the 8 channels P, R, S, supplying each 11 people must have an area of $11 \times 4 = 44$ square inches. The 7 channels D have to supply 110 people including P, R, S, or 16 each, and must be $16 \times 4 = 64$ square inches area. The eight exit openings Y and channel W should have an area of $1584 \div 8 = 200$ square inches each; but allowance should here be made for the bars of the ornamental grating by which the opening is covered and for contraction (153). With ·8 coefficient, each opening must have a *free* area of $200 \div \cdot 8 = 250$ square inches; the bars of the grating may probably occupy one-fifth of the total area, which should therefore be $250 \times 5 \div 4 = 312$ square inches.

The effect of substituting for the fan apparatus an ordinary ventilating chimney heated by coals or gas is shown by (386).

(373.) "*Hospitals.*"—The ventilation of a hospital must be more perfect, powerful, and uniform than any other; the state of health of the inmates necessitates a larger volume of fresh air than is necessary for persons in good health, and the ventilation must be continuous night and day without intermission. For perfect comfort, the *walls* should be at least as hot as the air in the room, which as we have seen (310) is impossible, where they have to be heated by the air in the room. Mechanical ventilation by a fan is not admissible, because in most cases hospitals are too large to be thus ventilated without an engine, to work which night and day is expensive; besides, as the work to be done cannot be remitted even for an hour, we

should require a duplicate set of apparatus in case of repairs. A chimney in which a draught is maintained by a fire is the best plan, because the mass of brickwork in the chimney retains so much heat as to maintain a fair draught for hours after the fire has been suffered to die out: see (397) and Table 116.

(374.) Figs. 109–112 represent a small hospital, or the wing of a large one, in which heated air is supplied in winter by a cockle or hot-air stove, and the ventilation is maintained at all seasons by a draught chimney. A is the hot-air apparatus, consisting of a collection of pipes B, open at both ends and built into the side walls, which are retained in position by clamp plates and bolts, the fire from the furnace at C circulates among the pipes in its passage to the chimney D, which serves for the escape of the smoke, &c., as well as the foul air from the rooms. The air to be heated enters the pipes at E, and passes through them into the chamber F, from which it proceeds by the channels G G, which run longitudinally the whole length of the building. Other channels H H are made in the main walls, by which the highly-heated air passes to the top of the building, thence descending by the channels J J to the bottom again, imparting the requisite heat to the walls in its passage. From the channels J J, branch pipes N N are laid, discharging the heated air (which by this time is cooled down to the proper temperature) into other long channels O O, and thence into the room by openings under each bed and at suitable places in the offices, &c., &c., in the ground floor.

(375.) The air thus received into the rooms passes through and ventilates them, escaping by orifices K K, &c., in the ceiling, into foul-air channels L L, which conduct it to the end of the building, where it enters the descending shafts M M, which communicate with the chimney D, by which it is finally discharged into the atmosphere.

The heated air enters the walls at a much higher temperature than is desirable for the rooms, but the walls absorb its surplus heat, and become heated perfectly throughout their mass, the air having for the most part to traverse the height of the building twice, before it escapes into the rooms. The direction of the currents is shown by arrows, and it will be observed that

the vitiated air moves off directly into the atmosphere, without mixing with the rest of the air in the room.

(376.) "*Loss of Heat by the Building, &c.*"—Allowing 2000 cubic feet of air per head per hour—see (331) and Table 112—and 150 invalids, we shall require 300,000 cubic feet, or $300000 \times \cdot 0761 = 22830$ lbs. of air per hour. If we assume that the internal surface of the wall is at 60° and the external *air* on a winter's day at 30°, with a thickness of 27 inches, we shall have (299) by the formula $U = \dfrac{Q \times (t - T')}{1 + Q\dfrac{E}{C}}$, or in our case $\dfrac{1 \cdot 134 \times (60 - 30)}{1 + \left(1 \cdot 134) \times \dfrac{27}{4 \cdot 83}\right)} = 4 \cdot 63$ units per square foot per hour, and as we have 11,790 square feet of wall surface (windows excepted), the heat dissipated by them will be $11790 \times 4 \cdot 63 = 54567$ units per hour. This heat has to be supplied by the air before it enters the building, and to do that it must be cooled $54567 \div (22830 \times \cdot 238) = 10°$, and as it leaves the walls to enter the rooms at 60°, it must enter them at 70°.

(377.) "*Cockle or Air-stove.*"—We require by (376), 22,830 lbs. of air per hour for ventilation, which air has to be heated 40°, or from 30° to 70°, and will require $22830 \times 40 \times \cdot 238 = 217340$ units of heat. With a cockle such as A in Fig. 112 we should not expect more than 6000 units usefully from a pound of coal (112), hence we require $217340 \div 6000 = 36$ lbs. of coal per hour, and $36 \div 12 = 3$ square feet of fire-grate (127). This, however, is the maximum consumption for the coldest weather, for which of course the heating apparatus must be adapted. By (346) the *mean* consumption during the six cold months of the year would be reduced to $36 \times 18 \div 30 = 22$ lbs. per hour, or $22 \times 24 \times 30 \times 6 \div 2240 = 42$ tons for that season.

In fixing the sizes of the heating pipes, two things must be considered: the bore must be such as to give the necessary area for the passage of the air, and the surface such as is necessary to heat that air in its passage through them. We have 300,000 cubic feet of air per hour, or 83 cubic feet per second,

and the velocity given by the chimney (379) being 7 feet per second, the combined area of the pipes must be $83 \div 7 = 12$ square feet.

(378.) A pipe heated externally as in our case gives out no *radiant* heat to the air within (245), but heats that air only by contact. For such a case it is advisable to use for the fire a larger volume of air than is absolutely necessary to effect combustion, in order to keep down the temperature as it leaves the furnace. By Table 45, if we used only the normal quantity, the air would have the high temperature of 2256°, approaching a white heat by Table 32, which would be destructive to the pipes, and injurious to the air passing through them. With a double volume of air, or 44·8 lbs. per pound of coal, the temperature would be reduced to 1159°; before it leaves the cockle this air parts with 217,340 units (377) in warming the air for ventilation, and as with 36 lbs. of coal per hour, the maximum winter consumption (377), we have $44 \cdot 8 \times 36 = 1613$ lbs. of air, the temperature will be reduced $217340 \div (1613 \times \cdot 238) = 566°$, and will become $1159 - 566 = 593°$ as it leaves the cockle. The mean temperature of the pipes is therefore $(1159 + 593) \div 2 = 876°$, and the mean temperature of the air passing through them being $(30+70) \div 2 = 50°$, the difference is $876 - 50 = 826°$, the ratio for which by Table 105 is about 2·3, and taking as an approximation the value of A from Table 99 at ·5, we have $\cdot 5 \times 2 \cdot 3 \times 876 = 1007$ units per square foot per hour, and the *internal* area must be $217340 \div 1007 = 216$ square feet. But with pipes arranged on one another as in Fig. 112, the full external surface is not effectively exposed to the heat, probably not more than two-thirds can be reckoned on, hence we require $216 \times 3 \div 2 = 324$ square feet.

The sizes of the pipes must therefore be such as to give an internal surface of 324 square feet, the corresponding external surface being exposed to the fire; and a cross-sectional area of 12 square feet. These conditions are nearly fulfilled by pipes 9½ inches bore. The cross-sectional area of 9½ inches is ·5 square foot, and with twenty-five pipes we have $\cdot 5 \times 25 = 12 \cdot 5$ square feet. With pipes 9 feet long, allowing that 18 inches at each end is lost by being built into the side walls,

the effective length is reduced to 6 feet, and the circumference of $9\frac{1}{2}$ inches being $2 \cdot 5$ feet, we have $2 \cdot 5 \times 6 \times 25 = 375$ square feet, or rather more than we require.

(379.) "*Ventilation.*"—The proportions of the ventilating apparatus must be fixed with particular reference to summer requirements when a fire has to be maintained specially for that purpose. For summer ventilation the damper P is closed, and a fire is maintained night and day (373) in the furnace R at the base of the chimney: this has a closed ash-pit, so that the air feeding the fire is drawn from the body of foul air that has passed through the wards and is on its way to the chimney. The external air of a summer day may be at 72°, and being heated, say 50°, or to 122°, we have in the chimney a column of air at 122° and 60 feet high, which by Table 24 would require a column of external air at 72° equal to $60 \times \cdot 0694 \div 0747 = 55 \cdot 74$ feet high, giving $60 - 55 \cdot 74 = 4 \cdot 26$ feet head to produce motion, which, by the laws of falling bodies, will generate a velocity of $\sqrt{4 \cdot 26} \times 8 = 16 \cdot 5$ feet per second theoretically (163). Admitting the ratio of the real to the theoretical velocity given by the experiments at the Prison Mazas (394), this is reduced to $16 \cdot 5 \times \cdot 423 = 7$ feet per second, and as we have to pass $300000 \div 3600 = 83$ cubic feet per second, the chimney and main air-passages must have an area of $83 \div 7 = 12$ square feet, or 1728 square inches, and must be 3 feet 6 inches square.

The area of the two channels G G will each be 6 square feet: $H = 1728 \div 18 = 96$ square inches; $N = 1728 \div (18 \times 3) = 32$ square inches; and $L = 1728 \div 6 = 288$ square inches, &c.

(380.) The cost of maintaining the ventilation in winter is *nothing*, the waste heat from the heating cockle being used; but in summer we have to heat the air required for ventilation 50° or from 72° to 122°, and shall require $(22830 \times 50 \times \cdot 238) \div 12000 = 22$ lbs. of coal per hour, and this will be uniform or nearly so throughout the six months of summer. We have therefore $22 \times 24 \times 30 \times 6 \div 2240 = 42$ tons, or precisely the same as for heating, &c., in the winter season (377).

It is apparently anomalous that for ventilation only, and when moreover nearly the whole of the heat in the fuel is utilized, we require the same amount of fuel as for heating and ventilation

in winter with an apparatus in which only *half* the total heat in the fuel is utilized. But it must be observed, that although only half the heat is utilized so far as warming the air for ventilation is concerned, the other half is really utilized in effecting the ventilation by heating the air in the chimney, which otherwise would have had only the low temperature of 60° at which it enters from the building. Thus in both cases the whole of the heat in the fuel (radiation excepted) is eventually given to the air in the chimney, the difference being that in winter the heated air is caused to pass through the building, and in summer it is not.

If there were no ventilation by the chimney whatever, a fan or some other method being adopted for effecting it, the cockle would still require the same amount of fuel merely to heat the air, so that the winter ventilation costs nothing practically, as already stated (230).

The ventilation in summer can be regulated by the consumption of fuel in the furnace R, and in winter by dampers S S in the main channels G G.

(381.) "*Cubic Capacity of Wards.*"—Medical men agree in demanding a large cubic capacity in the wards of hospitals, irrespective of ventilation or change of the air. 1200 cubic feet per bed has been given as the minimum for this country, and 1500 for tropical climates. Twenty-five English hospitals (ten being metropolitan and fifteen provincial) gave a mean of 1340 cubic feet, ranging from 988 at Nottingham to 2400 at the Royal Free Hospital, London. In our case, Fig. 109, we had $(100 \times 50 \times 13) \div 48 = 1354$ cubic feet per bed.

(382.) "*Effect of different Combustibles in Ventilation.*"—The effect of different kinds of fuel in creating a draught for ventilation has been made the subject of experiments by M. Morin, and the results are given by Table 116. They were made on an ordinary open fire-place, but apply with certain modifications to cases on the large scale.

The chimney of an ordinary fire-place serves not only to carry off the smoke, &c., but gives also a powerful ventilation, the air of the room being drawn in by the open throat above the fire. Nor is it only when a fire is used that ventilation is thus

effected, usually the air in the chimney without a fire is warmer or colder than the external air, and in either case there will be draught, upwards in one case, and downwards in the other. In summer time our dwellings are usually cooler than the external air, and the chimney being cooler also, a down-draught will be established; but in winter this is reversed; the internal air of the room is frequently 20° to 30° warmer than the external air, and that in the chimney taking nearly the same temperature, a powerful upward draught is created, as shown by Table 116; with an area of 2·31 square feet in the chimney so large a volume as 14,232 cubic feet of air per hour was discharged without any fire whatever; allowing 250 cubic feet per head (331), this would suffice for $14232 \div 250 = 57$ persons. With a moderate consumption of coals or wood about three times this volume was discharged, or sufficient for 170 persons.

(383.) "*Effect of Coals.*"—By col. 10, the mean useful result per pound of coal was 11,857 units, and as by (60) the total heat in coals is 13,000 units, $11857 \div 13000 = \cdot 91$, or 91 per cent. passes off by the chimney, and only 9 per cent. is dissipated by radiation to the walls of the room, &c. It would appear at first sight that 9 per cent. is all the heat usefully realized in an ordinary open fire. This, however, would be incorrect, for part of the heat in the air within the chimney has really been usefully employed in heating the air in the room before it entered the chimney. The useful effect of coals for *ventilation* purposes may be taken at 12,000 units per pound.

(384.) "*Effect of Wood.*"—By col. 10 the mean useful result of a pound of wood was 6722 units, but as by Table 44, the *total* heat in perfectly dry wood does not exceed 6582 units, there would appear to be some error of observation. By col. 2 of Table 44 the heat dissipated by radiation under ordinary circumstances is less with wood than with coals in the ratio of 23 to 50. If we admit 5 per cent. of the total heat in wood to be thus dissipated, that in air in the chimney will be $6582 \times \cdot 95 = 6253$ units per pound of perfectly dry wood, and $5265 \times \cdot 95 = 5000$ units per pound of wood in the ordinary state of dryness: we have here taken the full theoretical values given by Table 44.

TABLE 116.—Of the VENTILATION produced in an ORDINARY CHIMNEY by different COMBUSTIBLES consumed in the FIRE-PLACE as usual; also the effect of 106 Gas-jets placed within the Chimney. From MORIN'S Experiments.

Kind of Fuel.	Temperature of the Air. External, T.	Temperature of the Air. In the Room, t.	Temperature of the Air. In the Chimney, t'.	Volume of Air drawn by the Chimney in Cubic Feet per Hour, at Temp. Col. 3. Due to Natural Ventilation.	Due to the Combustible.	Total.	Increase of Temp. of the Air, $t'-t$.	Fuel Burnt per Hour.	Units of Heat Utilized per lb. of Fuel.	Volume of Air evacuated per lb. of Fuel.	Velocity of Air in the Chimney in Feet per Second.
	°	°	°				°	lbs.		cubic feet.	
No Fire	51·4	71·6	71·6	12609	..	12609	0	..	..	..	1·51
"	34·2	59·0	64·4	15753	..	15753	..	..	..	..	1·89
"	50·0	66·2	64·4	14234	..	14234	..	..	..	..	2·49
Mean	..	..	..	14232							
Wood Fire	59·0	66·2	194·0	14127	30658	44785	127·8	17·3	5902	1760	16·53
"	46·4	64·4	224·6	14127	34790	48212	160·2	18·4	7542	2608	16·63
Coal Fire	..	71·6	215·6	14127	27480	41607	144·0	10·3	10363	2570	15·68
"	..	69·8	190·4	14127	27409	41536	120·6	7·81	11410	3455	15·12
"	..	81·6	176·0	14127	32353	46480	94·4	6·27	13367	5136	16·50
"	..	68·0	264·2	14127	28715	42842	196·2	9·28	11452	3028	16·92
"	59·0	68·0	189·4	14127	29315	43442	121·4	9·20	12692	3168	18·14
"	65·3	68·0	107 6	14127	12302	26429	39·6	4·4	..	4384	8·46
								Cubic Feet.	Per Cubic Foot.	Per Cubic Foot of Gas.	
Gas	59·0	64·4	154·4	6746	29492	36238	90·0	83·2	697	430·9	12·6
"	68·0	66·5	185·0	6746	34932	41678	118·5	138·8	641	300	15·2
"	72·5	68·0	185·0	6746	32988	39734	117·0	141·3	587	281	14·4
"	60·8	68·0	78·8	6746	13916	20662	18·0	7·7	504	1818	6·3
"	63·5	67·1	82·4	6746	21262	28008	18·9	11·76	633	1808	8·7
"	62·6	65·3	113·0	6746	20733	27479	50·4	34·15	634	607	8·92
"	59·0	67·2	132·0	6746	27232	33978	81·9	70·64	504	385	11·35
"	64·4	67·2	149·0	6746	29845	36591	84·6	88·30	514	378	9·60
"	64·0	68·0	150·8	6746	30870	37616	86·8	93·1	516	332	12·95
"	64·0	68·0	167·0	6746	31894	38640	103·0	106·0	542	301	13·64
"	64·4	67·1	143·1	6746	33130	39876	101·7	122·8	611	269	14·23
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)

(385.) "*Effect of Gas.*"—The mean useful effect of a cubic foot of coal-gas, as given by col. 10 of Table 116, is 580 units, and is practically the same throughout, although the consumption was purposely varied for the sake of experiment from 7·7 to 141·3 cubic feet of gas per hour. But the practically useful result as estimated by the volume of air discharged *due to the action of the gas*, and shown by col. 11, varies exceedingly, partly as a natural consequence of known laws governing the question (150), and partly because there was a considerable ventilation without any gas whatever; with discharges of air in the ratio 1, 2, 3, 4, the consumption of gas would be about in the ratio 1, 14, 45, 63. The economy of small consumption of gas and consequent low temperature in the chimney, is thus shown very conclusively.

(386.) The ventilation of chapels, &c., which are usually occupied only about three or four hours per week, may be conveniently and comparatively cheaply effected in summer time by a draught-chimney in which the air is heated by gas. Say we take the case of the chapel in (359) with 400 people, requiring by (331) 500 cubic feet per head, or 200,000 cubic feet per hour, with a chimney 30 feet high, and an external summer temperature of 72°. If we assume that the air in the chimney shall be heated 30°, or to 102°, then by Table 24 a 30-foot column of air at 102° balances only $30 \times 0707 \div 0747 = 28 \cdot 4$ feet of external air at 72°; hence we have $30 - 28 \cdot 4 = 1 \cdot 6$ foot of unbalanced pressure, which will generate theoretically a velocity of $\sqrt{1 \cdot 6} \times 8 = 10$ feet per second; admitting as before (343) that by contractions, &c., this is reduced to half, we have 5 feet per second, the same as allowed in the mechanical ventilation. Now, 200,000 cubic feet of air at 72° is dilated to $200000 \times 1 \cdot 143 \div 1 \cdot 082 = 211300$ at 102°, or $211300 \div 3600 = 58$ cubic feet per second, hence the area of the draught-chimney must be $58 \div 5 = 11 \cdot 6$ square feet, or 3 feet 5 inches square.

The weight of air discharged per hour is $200000 \times \cdot 0747 = 14940$ lbs., to raise the temperature of which 30° requires $14940 \times 30 \times \cdot 238 = 101672$ units of heat, of which the audience will supply $191 \times 400 = 76400$, leaving $101672 - 76400 = 25272$ units to be supplied by the gas. Admitting, as in (385)

Morin's experiments, 580 units per cubic foot of gas, we shall consume $25272 \div 580 = 44$ cubic feet of gas per hour.

With half a congregation (368) we require only $101672 \div 2 = 50836$ units per hour to heat the air; and as under those circumstances each person emits only 127 units to the air, we have $127 \times 200 = 25400$ units from the audience, leaving $50836 - 25400 = 25436$ units to be supplied by the gas, or about the same as with a full congregation. In either case much of the heat required to effect the ventilation is supplied by the audience itself.

(387.) "*Heating Buildings by Gas.*"—Gas offers considerable advantages for heating purposes, being completely under control, and requiring no skilful management or constant attention, such as is necessary with hot-water or steam pipes; it is, however, considerably more costly. From the experiments of Péclet on the flame of an oil lamp, with the apparatus described in (90), 18 per cent. of the total heat is given out by radiation, and 82 per cent. to the air supporting combustion. Morin estimated the heat given out to the air by the flame of coal-gas in his experiments, Table 116, at five-sixths, or 84 per cent., 16 per cent. being absorbed and dissipated by the walls of the chimney. From this we find the *total* heat to be $580 \times 6 \div 5 = 696$ units per cubic foot.

(388.) "*Relative Cost of Coals and Gas for Heating and Ventilating Buildings.*"—Before we can compare the relative cost of coals and gas, we must consider the particular circumstances under which they are consumed. In a ventilating chimney nearly the whole of the heat is utilized, or say 12,000 units per pound of coal, and 580 units per cubic foot of gas: then taking the price of coals at 25*s*. per ton, we have $12000 \times 2240 \div (25 \times 12) = 89600$ units for one penny; with gas at 4*s*. per 1000 cubic feet we have $580 \times 1000 \div (4 \times 12) = 12100$ units for a penny; the ratio is $89600 \div 12100 = 7 \cdot 4$ to 1 for *ventilation.*

(389.) But for heating purposes we have frequently to make use of a steam or hot-water boiler, with which there are unavoidable losses from radiation and escape of heated air by the chimney, by which the heat really utilized is very much reduced,

as shown by (103) and Table 47. With small boilers, such as are used for ordinary cases, the losses are greater than the average, and we cannot reckon on more than from 4000 to 6000 units per pound of coal (112); allowing 6000, or half the amount in (388), we have 44,800 units for a penny. With gas, on the contrary, the whole of the heat is given out usefully either to the air, or to the walls of the room in which the gas is consumed, and we then have $696 \times 1000 \div (4 \times 12) = 14500$ units for a penny, and the ratio becomes $44800 \div 14500 = 3 \cdot 09$ to 1

But, if a stove with a very long flue-pipe be admissible, we may obtain $94\frac{1}{2}$ per cent. of the total heat in the coals (250), or $13000 \times \cdot 945 = 12285$ units per pound. In that case we have $12285 \times 2240 \div (25 \times 12) = 91728$ units for a penny, and the ratio is $91728 \div 14500 = 6 \cdot 3$ to 1.

(390.) Paraffine oil or petroleum may be used with advantage for ventilation, or heating in many cases on a small scale. By (59) the total heat is 20,240 units per pound; a gallon weighs 8·4 lbs., and costing say 18 pence, we have $20240 \times 8 \cdot 4 \div 18 = 9445$ units for a penny when it is consumed in the room, and the whole of the heat is utilized by being given out either to the walls or to the air (83); the ratio to gas, consumed under the same circumstances, is $14500 \div 9445 = 1 \cdot 53$ to 1.

As applied to a ventilating chimney where five-sixths of the total heat is realized, we have for petroleum $9445 \times 5 \div 6 = 7871$ units for a penny, and the relative cost of coals, gas, and petroleum is 89,600, 12,100, and 7871 units for a penny respectively, the ratio being 1, 7·4, and 11·4.

As applied to heating buildings where the coals are consumed under a boiler, and the gas and petroleum are burnt in the room, we have for coals, gas, and petroleum, 44,800, 14,500, and 9445 units for a penny respectively, the ratio being 1, 3·09, and 4·74.

But as applied to heating buildings, where the coals are burnt in a stove with a long flue-pipe, and the gas and paraffine are consumed in the room, we have 91,728, 14,500, and 9445 units for a penny respectively, and the ratio is 1, 6·3, and 9·7.

CHAPTER XV.

EXAMPLES OF BUILDINGS HEATED AND VENTILATED.

As illustrations of the application of the rules given in Chapter XII. for calculating the loss of heat by buildings, and to show how far they agree with observed facts, we may take the following examples, the data for which are given by Péclet.

(391.) "*Prison Mazas.*"—This great prison is situated in Paris, and contains cells for the solitary confinement of 1200 prisoners. These cells are arranged in six long buildings, which radiate from a centre, and occupy nearly two-thirds of a circle. The walls are of stone, about 36 feet high, averaging 24 inches thick, and exposing an exterior area of 140,000 square feet; the area of the windows is 23,400 square feet. For the seven months of the year, during which the building was heated, the mean temperature of the exterior air was 44°, and the interior air 58°, being a difference of 14°, and from these particulars we can calculate the weight of coals required by the building, &c. The ventilation was maintained constantly by a chimney with a fire at its base, as in Fig. 112; this chimney was cylindrical, 7 feet internal diameter and 95 feet high, the mean quantity of air passing through the building was 1,059,600 cubic feet per hour, and this air had to be heated 14°, from the external to the internal temperatures, and required a further quantity of fuel.

(392.) The conditions of the building are similar to those in (302) and Fig. 84, with one face exposed, and by Table 103 we find that a wall of stone 24 inches thick loses ·284 unit per square foot for 1° difference of internal and external temperature of the air. In our case, therefore, the walls will lose $\cdot 284 \times 14 \times 140000 = 556640$ units per hour. The windows were for the most part only 2 feet high, and by the rule in (305) lose ·56 unit per square foot for 1°. The loss in our case is $\cdot 56 \times 14 \times 23400 = 183456$ units per hour.

By Table 24 the weight of air at say 42° is ·079 lb. per cubic foot; we have therefore $1059600 \times \cdot 079 = 83708$ lbs.

of air to be heated 14°, and the specific heat of air being ·238, this is equal to $83708 \times \cdot 238 \times 14 = 278910$ units per hour. Collecting these three results, we have a total loss of $556640 + 183456 + 278910 = 1019006$ units per hour. But part of this heat will be supplied by the animal heat of the prisoners, as explained in (327), where we find that each man will yield 191 units per hour, and we have $191 \times 1200 = 229200$ units per hour from this source, leaving $1019006 - 229200 = 789806$ units to be supplied by the heating apparatus. Allowing 6000 units per pound of coal (112), we should require $789806 \times 24 \div 6000 = 3160$ lbs. of coal per day. The experimental quantity was 3564 lbs., showing that only 5318 units per pound of coal were utilized, or $5318 \div 13000 = \cdot 41$, or 41 per cent. of the *total* heat in coals (60).

(393.) During the cold weather of winter it was found that 5280 lbs. of coal were consumed per 24 hours, to maintain the internal air at 59°·3 while the external air was 39°, the difference being 20°·3. We found that for 14° difference the loss was 1,019,006 units; the loss with 20·3 will therefore be $1019006 \times 20 \cdot 3 \div 14 = 1477560$, and deducting the heat emitted by the prisoners, we have $1477560 - 229200 = 124836$ units per hour to be supplied by the fuel. In this case, therefore, $124836 \times 24 \div 5280 = 5675$ units per pound of coal were utilized, or nearly the same as in the case of the Church of St. Roch (400).

(394.) "*Ventilation.*"—For maintaining the ventilation it was found that the mean consumption of coals by the fire at the foot of the chimney was 770 lbs. per day in winter, and 880 lbs. per day for the rest of the year; but for the ventilation, equal to 1,059,600 cubic feet per hour, the consumption was 44 lbs. per hour in winter, and 55 lbs. in summer.

The temperature of the air in the chimney was not observed, but we can calculate it from the consumption of fuel. By (60) the *total* heat in a pound of coals is 13,000 units; nearly the whole of this will be given out to the air passing up the chimney; allowing that 5 per cent. falls from the grate unconsumed as in (98), and that 5 per cent. more is lost by radiation, &c., we have from 44 lbs. of coals $13000 \times \cdot 9 \times 44 = 514800$

units of heat per hour, and as this is carried off by 83,708 lbs. of air, and the specific heat of air being $\cdot 238$, we have $\frac{514800}{\cdot 238 \times 83708} = 26°$ increase of temperature, and the air entering at 58° (the temperature of the building) becomes $58° + 26° = 84°$ in the chimney, or $84 - 44 = 40°$ warmer than the exterior air. A 95-foot column of air at 84° is equal by (27) to $95 \times \frac{458 \cdot 4 + 44}{458 \cdot 4 + 84} = 88$-foot column of air at 44°, and we have $95 - 88 = 7$ feet head to generate velocity, which will give $\sqrt{7} \times 8 = 21 \cdot 1$ feet per second. The chimney was 7 feet diameter, having an area of $38 \cdot 5$ square feet; but the centre of it was occupied by the iron chimney from the steam-boilers, which was 2 feet 8 inches diameter, having an area of $5 \cdot 5$ square feet. The acting area of the air-shaft was thus reduced to $38 \cdot 5 - 5 \cdot 5 = 33$ square feet, and when discharging 1,059,600 cubic feet per hour, the velocity of the air entering the chimney must be $\frac{1059600}{83 \times 60 \times 60} = 8 \cdot 92$ feet per second. The real velocity is therefore only $8 \cdot 92 \div 21 \cdot 1 = \cdot 423$ of the theoretical velocity, this loss arising from friction, change of velocity by frequent contractions and enlargements in the air-channels, &c., &c., which are practically unavoidable in long circuits: see (163). The mechanical work done by the chimney being proportional to the square of the velocity, is thus reduced to $\cdot 423^2 = \cdot 179$.

(395.) "*Mechanical and Chimney Ventilation Compared.*"—We may compare mechanical ventilation with that produced by heat, and ascertain the relative economy of the two systems, by calculating the power of an engine and fan capable of doing the same work and the consumption of fuel in the two cases.

By (394) we have seen that with the standard quantity of air the consumption of coals was 44 lbs. per hour during the five cold months of the year, and 55 lbs. per hour for seven months. We have therefore $(44 \times 24 \times 30 \times 5) + (55 \times 24 \times 30 \times 7) = 435600$ lbs. of coals per year, with a heated chimney.

With a high-pressure engine the work in winter costs *nothing*, because the steam after working the engine is used for heating the building (230).

The mechanical work done is 83,708 lbs. of air per hour, or 1228 lbs. per minute, at a velocity of 8·92 feet per second, the head due to which, by the law of falling bodies, is $\left(\frac{8 \cdot 92}{8}\right)^2 =$ 1·243 foot; we have therefore $1228 \times 1{\cdot}243 \div 33000 = {\cdot}0463$ horse-power. But we have seen that by friction, &c., &c., in the passages the work is reduced to ·179 of the power employed; and as a low-pressure fan and gearing will probably yield only one-third of the force expended on it, we should require ${\cdot}0463 \times 3 \div {\cdot}179 = {\cdot}776$ horse-power. If we allow 10 lbs. of coal per horse-power, this is equal to 7·76 lbs. of coal per hour for seven months of the year only, and the yearly consumption would be $7{\cdot}76 \times 24 \times 30 \times 7 = 39110$ lbs. of coal, instead of 435,600 lbs. as required by a draught chimney; the ratio is 1 to 11.

This is a favourable case for mechanical ventilation, being a very large one. By the ordinary allowance of 350 cubic feet per head (331), the air dealt with would suffice for $1059600 \div 350 = 3000$ persons, and even for this large number we require an engine of only three-fourths of a horse-power. For small, and even for ordinary cases, the engine would be excessively small, the friction proportionally much greater, and the relative economy of the system less. There is also the practical objection that the engine must work day and night, requiring an extra man for the night work, &c., and a duplicate engine in case of repairs, &c.

(396.) "*Prison of Provins.*"—This prison was arranged in the same manner as the Prison Mazas, but was very much smaller, consisting of one range of buildings for 39 prisoners. The walls were of stone, 24 inches thick, and exposed 11,340 square feet of surface. The windows had an area of 1157 square feet, and the air used for ventilation was 120,090 cubic feet per hour.

Experiments were made on the consumption of fuel from the 15th of March to the 6th of April, the mean temperature of the *day* was 43° by observation, that of the night was not observed, but is reported to have been very cold, probably it was 8° colder than the day, or 35°; the *mean* temperature of the exterior air

would therefore be 39°, and the internal air being maintained at 59°, the difference would be 20°. The consumption of fuel under these circumstances was found to be 807 lbs. of peat per day.

The loss by the walls 24 inches thick will be $\cdot 284$ unit for 1° by Table 103, and we have $\cdot 284 \times 20 \times 11340 = 64411$ units per hour lost by the walls. The windows by (392) will lose $\cdot 56 \times 20 \times 1157 = 12958$ units, and the air $120090 \times \cdot 079 \times \cdot 238 \times 20 = 45348$ units; collecting these three losses, we have $64411 + 12958 + 45348 = 122717$ units per hour. The prisoners will yield $191 \times 39 = 7449$ units, leaving 115,268 to be supplied by the fuel.

By (60) and (62) we find the *total* heat in coals and peat to be 13,000 and 7151 units respectively, and the economic value of coals being as we have assumed 6000 units, that of peat will be $6000 \times 7151 \div 13000 = 3300$ units per pound, and thus we shall require $115268 \times 24 \div 3300 = 838$ lbs. of peat, whereas experiment gave 807 lbs. We had therefore in this case $115268 \times 24 \div 807 = 3428$ units per pound of peat, or $3428 \div 7151 = \cdot 48$, or 48 per cent. of the total heat in the fuel.

(397.) "*Ventilation.*"—The draught or air-chimney was 59 feet high, and 2 feet diameter at the top, its area being $3 \cdot 14$ square feet. An experiment was made when all the fires had been extinguished six hours, the temperature of the air in the chimney would be about the same as that in the building, which was 16° higher than the external air, which being taken at 39°, that in the chimney would be 55°. The observed discharge of air under these circumstances was 40,052 cubic feet per hour, and the velocity of discharge must therefore be $\frac{40052}{3600 \times 3 \cdot 14} = 3 \cdot 54$ feet per second. For calculating the theoretical velocity we have a column of air 59 feet high at 55°, which is equal in weight to a column $59 \times \frac{458 \cdot 4 + 39}{458 \cdot 4 + 55} = 57 \cdot 11$ feet high at 39°, and we have $59 - 57 \cdot 11 = 1 \cdot 89$ foot head to generate velocity, which will give $\sqrt{1 \cdot 89} \times 8 = 11$ feet per second, whereas the real velocity was only $3 \cdot 54$ feet per second. The ratio in this case is $3 \cdot 54 \div 11 = \cdot 322$ to 1, and the mechanical work

of the chimney, only $\cdot 322^2 = \cdot 104$ of its theoretical value; see (394).

(398.) "*Church of St. Roch.*"—This church was about 377 feet long, 92 feet wide, and from 50 to 60 feet high. The walls were of stone, 20 inches thick, exposing a surface of 37,674 square feet; the windows were about 13 feet high, and exposed an area of 9257 square feet. When the interior air was maintained at 29° above the exterior air, the consumption of coals was 88 lbs. per hour.

The conditions of this building are similar to Fig. 80, the walls being exposed on all sides to cooling influences. The loss of heat for this case is given by Table 102, which for stone walls 20 inches thick may be taken at $\cdot 2$ for 1°; in our case the loss will be $\cdot 2 \times 29 \times 37674 = 218510$ units per hour. By (307) the windows lose $\cdot 4$ unit for 1°, or in our case $4 \times 29 \times 9257 = 107385$ units, and the loss from both sources is 325,895 units per hour. There will also be another loss of heat by a large but unknown volume of air, which enters through the heating apparatus by openings in the floor, and passes out by innumerable crevices in the leaden casements, the glass being very loosely fitted; it will therefore escape at the temperature of the glass, and if we know that temperature we can calculate the volume of air which passes through the building.

(399.) "*Temperature of the Walls and Glass in Windows.*"—To give distinctness to the question, we will assume that the internal air is at 59°, and the external air at 30°, or 29° difference. We can now calculate the temperatures of the two surfaces of the walls by the rules in (293) and (294). Taking the value of R from Table 95 at $\cdot 736$, of A from Table 97 at $\cdot 398$, and of C from Table 101 at $13 \cdot 7$, Q will be $1 \cdot 134$, and the rule in (293), or

$$\frac{\left\{Q \times [E \times A \times T) + (C \times T]\right\} + \left\{A \times C \times T\right\}}{\left\{C \times [2 \times A) + R]\right\} + \left\{E \times A \times Q\right\}} = t$$

becomes

$$\frac{\left\{1 \cdot 134 \times [20 \times \cdot 398 \times 59) + (13 \cdot 7 \times 30]\right\} + \left\{\cdot 398 \times 13 \cdot 7 \times 59\right\}}{\left\{13 \cdot 7 \times [2 \times \cdot 398) + \cdot 736]\right\} + \left\{20 \times \cdot 398 \times 1 \cdot 134\right\}} = 44°$$

which is the temperature of the interior surface of the wall. The temperature of the exterior surface may now be found by the rule in (294), or $\frac{C\,t + Q\,E\,T'}{C + Q\,E} = t'$, which in our case becomes $\frac{(13 \cdot 7 \times 44) + (1 \cdot 134 \times 20 \times 30)}{13 \cdot 7 + (1 \cdot 134 \times 20)} = 35^\circ \cdot 27$.

The temperature of the glass in the windows will be given by the rule in (307), $t'' = \left(\frac{T - t) \times A}{A + R} + t + T'\right) \div 2$, which in our case becomes $\left(\frac{59 - 44) \times \cdot 43}{\cdot 43 + \cdot 5948} + 44 + 30\right) \div 2 = 40^\circ \cdot 15$, and this is also the temperature of the air at exit.

(400.) "*Volume of Air.*"—We can now calculate the volume of air passing through the building, for as it departs at $40^\circ \cdot 15$, it must have been cooled by the walls from 59° to $40 \cdot 15 = 18^\circ \cdot 85$, and the weight of air which in doing that would yield to the walls and windows the required quantity of 325,895 units must be $325895 \div (18 \cdot 85 \times \cdot 238) = 70342$ lbs., or $70342 \div \cdot 0791 = 889300$ cubic feet at 42°. This air departing at $40^\circ \cdot 15$ or $10^\circ \cdot 15$ above the external air, will carry off $70342 \times 10 \cdot 15 \times \cdot 238 = 170000$ units of heat, which added to the amount lost by the walls and windows gives a total of $325895 + 170000 = 495895$ units per hour, requiring $495895 \div 6000 = 82 \cdot 6$ lbs. of coal, instead of 88 lbs. as per experiment. In this case, therefore, $495895 \div 88 = 5635$ units were utilized per pound of coal; at Mazas (393) we found 5675 units.

(401.) But we might have calculated approximately both the volume of air and its temperature at exit, and therefore of the glass in the windows, from the known quantity of coals consumed. 88 lbs. of coals give $88 \times 6000 = 528000$ units of heat, and as the *whole* of this heat is given out to the *air* as it passes the hot-water pipes and enters the building, and that air being heated 29°, its weight must be $528000 \div (29 \times \cdot 238) = 76500$ lbs. This air has to yield 325,895 units to the walls, &c., and to do that must be cooled down $325895 \div (76500 \times \cdot 238) = 17^\circ \cdot 9$; it therefore departs from the building at $59^\circ - 17^\circ \cdot 9 = 41^\circ \cdot 1$, and this is also the temperature of the glass. The volume of air and temperature of glass as thus calculated differ but little

from those found by the former method, which, no doubt, is the more correct of the two.

(402.) The capacity of the spacious church was about 1,130,000 cubic feet of air, which added to 889,300 due to ventilation, gives in an hour a total of 2,019,300 cubic feet, or 505 cubic feet per head for 4000 people. The ordinary numbers present are 2000 to 4000; on fête days 4000 to 6000, and on grand occasions 6000 to 8000.

(403.) "*Time required to Heat the Building to the Standard Temperature.*"—It was found that to heat the building to the standard temperature, or 29° above the external air, required eight days of continuous firing day and night, and it will be instructive to see how far this agrees with calculation.

The walls contain 63,576 cubic feet of stone, weighing by Table 37 about 156 lbs. per foot, or $63576 \times 156 = 9917856$ lbs., and the specific heat by Table 1 being about $\cdot 21585$, they will require $9917856 \times \cdot 21585 = 2140770$ units of heat to raise their temperature 1°. We found in (399) that when the standard temperature was attained, the two surfaces are 44° and $35° \cdot 27$ respectively, the *mean* temperature of the wall is therefore $(44 + 35 \cdot 27) \div 2 - 39° \cdot 625$, or $9° \cdot 625$ above the external temperature, and this is the amount of heat which the walls must receive before the standard temperature can be attained. We found that for 1° they required 2,140,770 units, they will therefore require $2140770 \times 9 \cdot 625 = 20626320$ units to bring them up to the standard.

(404.) The time in which this quantity can be supplied is in our case governed by the maximum power of the heating apparatus. This was proved with an external temperature of 21°, for in that case the internal temperature could not be maintained higher than 50°, being a difference of 29°, or the same as at the standard internal temperature of 59°, with external air at 30°. The apparatus could therefore yield only 218,510 units per hour *to the walls* as in (398), and to do that, the temperature of the walls must be 15° colder than the air, as we have seen in (399), where the respective temperatures were 59° and 44°. This difference would be constant throughout, so that the walls being 30° at the commencement, the internal air would be heated

by the apparatus to 45°, when further increase of temperature would be arrested by the walls absorbing the heat, and as their temperature increased, so would that of the air be progressively and simultaneously increased, and the difference of 15° would be maintained throughout.

(405.) But while heat was thus received by the internal surface, a considerable amount would be dissipated by the external surface. At the commencement, when the walls had the same temperature as the external air, the loss would be 0, progressively increasing to 218,510 units per hour when the walls were heated to the standard temperature; for when that is attained, the external surface dissipates the same amount as the internal surface receives, and the temperature remains stationary. The *mean* loss is therefore $(218510 + 0) \div 2 = 109255$ units per hour, so that while 218,510 units were received by the interior surface, 109,255 units would be dissipated by the external surface, and 109,255 units would be retained by the walls; and to obtain the 20,626,320 units required to bring them up to the standard temperature (403) the time would be $20626320 \div (109255 \times 24) = 7 \cdot 87$ days; practically the same as by experiment, which, as we have stated (403), was eight days.

(406.) "*Time required to Cool the Building.*"—When once heated, a considerable time would be required to cool the whole building down to the external temperature. If the air passing through it be stopped by closing the inlet openings, heat would be dissipated only by the walls and windows, and we should lose 325,895 units at first, which would be progressively reduced to nothing at the end of the time. The mean loss would therefore be $(325895 + 0) \div 2 = 162947$ units per hour, and the heat of the building would be dissipated in $\frac{20626320}{162947 \times 24}$ $= 5 \cdot 27$ days.

If the circulation of air be permitted as usual, the mean rate of loss would be $(528000 + 0) \div 2 = 264000$ units per hour, and the building would cool down to the external temperature in $\frac{20626320}{264000 \times 24} = 3 \cdot 25$ days.

(407.) "*Heating Apparatus.*"—The heating apparatus consisted of a boiler about 12 horse-power, with hot-water pipes $5\frac{1}{2}$ and $4\frac{3}{4}$ inches diameter. They were laid in channels with brick sides, under the floor of the church, in the usual way. The total area of the hot-water pipes was 1774 square feet, they therefore gave out $495895 \div 1774 = 279\cdot 5$ units per square foot: see (400). The temperature of the water as it left the boiler was 248°, and as it returned 216°, its mean temperature was therefore 232^6, or $232^\circ - 29^\circ = 203^\circ$ above the external air.

(408.) The case of a pipe enclosed in a channel is quite different to that of a pipe freely exposed. When a pipe is enclosed, the walls receive radiant heat from it, and their temperature is raised until they give out to the air in contact the same amount as they are receiving from the pipe, when further increase of temperature is arrested, and it remains stationary. With brick walls and a steam-pipe at 232° the temperature of the walls would be 158°. At that temperature, being $232 - 158 = 74^\circ$ colder than the pipe, the ratio (313) of heat lost by radiation is, by Table 104, $1\cdot 9$, and the value of R from Table 95 being for bricks $\cdot 736$, the walls will receive $\cdot 736 \times 74 \times 1\cdot 9 = 103$ units per square foot per hour. Then by (315) a wall, say 2 feet high, will have $\cdot 528$ for the value of A by Table 97, and the difference of temperature of the air and of the wall being $158 - 30 = 128^\circ$, the ratio by Table 105 is $1\cdot 486$, and the loss by contact of air is $\cdot 528 \times 128 \times 1\cdot 486 = 100\cdot 3$ units per hour, nearly the same amount as was received by radiation, and showing that the temperature of 158° is very nearly correct: see (317).

(409.) A steam-pipe 5 inches diameter, heated to 232°, exposed to radiant walls at 158°, and to air at 30° (see 316), will lose $(\overset{\text{R.}}{\cdot 7} \times \overset{\text{Diff.}}{74^\circ} \times \overset{\text{Ratio.}}{1\cdot 9}) + (\overset{\text{A.}}{\cdot 544} \times \overset{\text{Diff.}}{202^\circ} \times \overset{\text{Ratio.}}{1\cdot 65}) = 279\cdot 7$ units per square foot per hour, which by an accidental coincidence is almost exactly the amount found by experience, which we have seen to be $279\cdot 5$ units.

If this same pipe had been freely exposed to air and radiant objects both at 30°, the difference of temperature would have

been 232° − 30° = 202° in both cases, and the loss $(\overset{\text{R.}}{\cdot 7} \times \overset{\text{Diff.}}{202} \times \overset{\text{Ratio.}}{1 \cdot 51}) + (\overset{\text{A.}}{\cdot 544} \times \overset{\text{Diff.}}{202} \times \overset{\text{Ratio.}}{1 \cdot 65}) = 394 \cdot 8$ units, and an enclosed pipe therefore gives only $279 \cdot 7 \div 394 \cdot 8 = \cdot 708$, or say 70 per cent. of the amount lost by a pipe freely exposed (317).

CHAPTER XVI.

WIND, FORCE OF, AND EFFECT ON VENTILATION, ETC.

(410.) "*Influence of the Wind in Ventilation, &c.*"—The effect of the wind on the draught-power of chimneys to furnaces, &c., is well known by experience. Let A B C, Fig. 13, be a tube freely exposed to the action of the wind, A B being vertical; if now the wind moves in the direction C to B, it will enter at C and pass out at A; but if the wind moves from B to C, it will enter at A and pass out at C.

It will be seen that this is analogous to the case of a boiler-house on a plane, with a chimney at one end, as in Fig. 15, the whole being freely exposed to the wind on all sides, and air admitted by openings at the end C; but here the air in the chimney is highly heated, and a powerful upward draught thereby created; still it will be retarded when the wind sets from B to C, and assisted and accelerated when in the contrary direction. Table 62 shows the draught-power of chimneys in calm air, in inches of water pressure, by which we can compare the power of the draught with that of the wind.

(411.) According to the experiments of Hutton, the force of wind at moderate velocities varies as the $2 \cdot 04$ power of the velocity; he also found that a plane 32 square inches in area, and moving at a velocity of 12 feet per second, experiences a resistance from the wind of $\cdot 841$ ounce. From these data we obtain the formulæ

$V^{2 \cdot 04} \times \cdot 001487 = P$ and $\dfrac{P}{\cdot 001487} = V^{2 \cdot 04}$, in which V

= the velocity in feet per second, and P = the pressure in pounds per square foot; Table 117 has been calculated by these rules.

TABLE 117.—Of the FORCE and VELOCITY of WIND, according to HUTTON'S Experiments.

Pressure.		Velocity.		Character of the Wind according to Rouse and Lind.
In lbs. per Square Foot.	In Inches of Water.	In Feet per Second.	In Miles. per Hour.	
·1	·01926	7·87	5·37	Gentle Wind.
·25	·0481	12·90	8·79	Pleasant ditto.
·5	·0963	16·18	11·03	Fresh Breeze.
1	·193	24·33	16·60	Brisk Gale.
2	·385	34·17	23·30	Very brisk ditto.
3	·578	41·69	27·77	
4	·770	48·00	32·73	
5	·963	53·55	36·51	High Wind.
6	1·155	58·55	39·92	
7	1·348	63·15	43·06	
8	1·541	67·43	45·97	
9	1·733	71·43	48·70	Very high ditto.
10	1·926	75·22	51·28	
11	2·118	78·82	53·74	
12	2·311	82·20	56·07	
13	2·504	85·54	58·32	
14	2·696	88·70	60·48	Storm or Tempest.
15	2·889	91·76	62·56	
16	3·081	94·70	64·57	
17	3·274	97·56	66·52	
18	3·467	100·30	68·41	
19	3·659	103·00	70·25	
20	3·852	105·70	72·03	Great Storm.
25	4·815	117·80	80·36	
30	5·778	128·9	87·97	Hurricane.
35	6·741	139·0	94·77	
40	7·704	148·7	101·4	Very great ditto.
45	8·667	157·2	107·2	
50	9·630	165·6	112·9	Most violent ditto.
55	10·593	177·5	118·3	
60	11·556	181·0	123·4	

(412.) Taking as an illustration a chimney 80 feet high, we find by Table 62 that the draught-power is equal to ·585 inch of water, and by Table 117 this is equal to a wind having a velocity of about 42 feet per second, which is a very brisk breeze, and if the boiler-house is freely exposed to its force, the

draught of the chimney would be destroyed when the wind was blowing with that strength in the direction B C, and *doubled* when in the direction C B. In such a case the doors or other openings at the end C should be closed, and others opened at the end B or at the side D.

(413.) To obviate the adverse action of the wind, and to utilize its power of increasing the draught, we may use a movable cowl, whose action and principle may be illustrated by Fig. 14, in which we have a tube like Fig. 13 surmounted by a cowl, movable round the centre of the tube A B by the action of the wind itself, which is assisted by the vane E. When the wind is in the adverse direction B C, the cowl A D opposes and balances the effect of the wind on the lower branch, and if both are equally exposed to its action, the one would simply neutralize the other, and there would be no internal current in either direction, but if, as is usually the case in practice, B C is sheltered by adjacent buildings, &c., and A D is fully exposed, the cowl will always be the controlling power, creating in all cases an upward current, irrespective of the action of heat on the internal air, and the adverse action of the wind is not only annulled, but reversed and made to augment the draught.

(414.) The effect of the vane E may be greatly increased by making it with double blades, as in Fig. 99. The experiments of Hutton show that the force of wind acting obliquely on a plane is given by the rule

$$P = V^{2\cdot 04} \times \cdot 001487 \times \text{sine}^{\,1\cdot 842\ \text{cosine}}$$

by which we find that the force at a right angle being 1·0, that with angles 1°, 2°, 3°, &c., is ·003, ·006, ·010, &c., as in Table 118. Hutton gives a table showing the effect at all angles from 1° to 90°, from which it will be found that the best angle for the blades is 38° with the axis, or 76° with each other, as in Fig. 100. To show the superior efficacy of a double-bladed vane, say that the wind was at an angle of 1° with the axis, then with a vane having blades at the best angle instead of impinging on both blades at 38°, we should evidently have it at 37° on one and 39° on the other, and by Hutton's table the

forces with these angles would be ·507 and ·555, and the force tending to turn the vane ·555 − ·507 = ·048 instead of ·003, as with a single blade of the same area, or ·048 ÷ ·003 = 16 times greater.

Table 118 shows that the superiority of the double blade over the single is greatest at small angles of wind, where in fact it is most wanted. Such a vane would not only be more sensitive, but also more steady, or less subject to oscillation. Angles greater or less than 38° are not so effective, the turning power at 15° would be about one-half, and at 5° about one-fifth of that at 38°.

TABLE 118.—Of the RELATIVE POWER of WIND-VANES, with Single and Double Blades.

	Angle of Wind with the Axis.							
	1°	2°	3°	4°	5°	10°	20°	30°
	Ratio of the Turning Force.							
Single Blade ..	·003	·006	·010	·014	·018	·046	·156	·347
Double Blades, angle 38° with axis	·048	·095	·141	·185	·228	·431	·739	·910
Ratio of effect, single blade being 1	16	15·8	14·1	13·2	12·7	9·4	4·7	2·6

(415.) "*Maximum Force of the Wind.*"—In this country the force of the wind seldom exceeds 50 lbs. per square foot even in our great storms; it has been known, however, occasionally to exceed that amount. In the storm of December 27, 1868, at Liverpool, the recorded pressure at 3.10 P.M. was 80 lbs. per square foot; equal to 142 miles per hour by the rule. The *mean* velocity observed between 3 and 4 P.M. was 92 miles per hour.

(416.) We will calculate the force of the wind capable of overturning the chimney 80 feet high, shown by Fig. 44. There are two forces which resist the wind, namely, the weight of the whole chimney and the cohesion of the mortar. The

chimney contains 1747 cubic feet of brickwork, which weighs, by Table 37, 115 lbs. per foot, giving a total weight of 200,000 lbs. The cohesion of good old mortar (fourteen years old) is 60 lbs. per square inch, and the area of the section of base being $81^2 - 36^2 = 5265$, the resistance of the cohesion of the mortar will be $5265 \times 60 = 315900$ lbs., which added to the weight of the chimney makes a total of $200000 + 315900 = 515900$ lbs. If the materials were incapable of crushing, the chimney would turn on that edge of its base remote from the wind; but in truth that point would be somewhere between the centre and the edge, and the chimney would resist fracture, partly by the crushing strain and partly by the cohesion assisted by the weight. By analogy with other materials broken transversely, we know that the result is very nearly the same as if the neutral axis coincided with the edge, and the force of cohesion only came into play. Admitting this, the force of 515,900 lbs. acts with a leverage equal to half the diameter of the base, or $40 \cdot 5$ inches. The centre of effort of the wind is at the centre of gravity of the surface exposed to it; the easiest way of finding the centre of gravity in our case is by *cutting out* an outline of the chimney in drawing paper, &c., of equable thickness, and balancing it on the point of a needle. We thus find the centre of effort in our case to be 36 feet, or 432 inches above the base, and the surface area of one side of the chimney being 440 square feet, the force of the wind that would overturn it would be $\frac{515900 \times 40 \cdot 5}{432 \times 440} = 110$ lbs. per square foot, which as we have seen is greater than any wind in this country. With very bad mortar, the force would be much less, for instance, without mortar altogether, the force of the wind would be $\frac{200000 \times 40 \cdot 5}{432 \times 440} = 42 \cdot 6$ lbs. per square foot.

This will show that a chimney is in the greatest danger when newly built, before the mortar has had time to set. For this and other reasons, it is expedient to proceed very slowly with the work of building: without this precaution, a chimney is apt to settle irregularly, and to become crooked and unsightly.

APPENDIX.

(417.) "*Explosive Force of Gases.*"—When combustible gases such as hydrogen and carbonic oxide, &c., are mixed with oxygen in the proportions necessary to effect combustion, and heat is applied by a match, &c., combustion follows instantaneously, or nearly so, and takes the form of an explosion. If we know the amount of heat developed by the combustible gas, and the volume and specific heat of the resulting product which receives that heat, we can show the temperature to which it will be raised by it, and thence the pressure or explosive force which would be generated in a closed vessel.

Say that we take the case of the explosion of carbonic oxide with oxygen: Carbonic oxide is composed of 1 atom, or say 75 lbs. of carbon (56), and 1 atom, or 100 lbs. of oxygen, thus forming 175 lbs. of carbonic oxide. To effect combustion another atom or 100 lbs. of oxygen must be added, and we then have before combustion 275 lbs. of mixed gases, and after combustion the same weight of carbonic acid gas.

Now by Table 39 the mixed gases before combustion occupied $(13 \cdot 6 \times 75) + (11 \cdot 88 \times 200) = 3376$ cubic feet at 62°, and this is the capacity of the closed vessel which would contain them. After combustion we have 275 lbs. of carbonic acid gas or $8 \cdot 59 \times 275 = 2362$ cubic feet at 62°.

The heat developed by the combustion of 1 lb. of carbonic oxide is 4325 units by the experiments of Favre and Silberman in Table 42; hence we have $4325 \times 175 = 756875$ units of heat, and the specific heat of carbonic acid gas with constant volume (6) being ·17112 by Table 5, the 275 lbs. would be heated $756875 \div (\cdot 17112 \times 275) = 16084°$, or to $16084 + 62 = 16146°$, at which temperature by the rule in (27) the 2362 cubic feet of carbonic acid gas would become $2362 \times \frac{458 + 16146}{458 + 62} = 75420$ cubic feet if expansion were permitted; but being confined in a vessel of 3396 cubic feet capacity, the pressure by the law of Marriotte (29) would become $75420 \div 3396 = 22 \cdot 2$ atmospheres.

With atmospheric air instead of pure oxygen the explosive force would be reduced. Atmospheric air is composed of 1 atom, or by Table 40, 100 lbs. of oxygen, and 2 atoms, or 350 lbs. of nitrogen; hence the *extra* 100 lbs. of oxygen required for the combustion is now associated with 350 lbs. of nitrogen

occupying $13 \cdot 54 \times 350 = 4739$ cubic feet, and the capacity of the closed vessel holding the gases before explosion would be $4739 + 3376 = 8115$ cubic feet.

After combustion we have as before 275 lbs. of carbonic acid, or 2362 cubic feet at 62°, which, added to the nitrogen, gives a total of $2362 + 4739 = 7101$ cubic feet at 62°. To heat the carbonic acid 1° we require $275 \times \cdot 17112 = 47$ units; to heat the nitrogen 1° we require $350 \times \cdot 17272 = 60$ units; hence to heat the products of combustion 1° we require $47 + 60 = 107$ units, and the 756,875 units developed by the combustion will raise their temperature $756875 \div 107 = 7073°$, or to $7073 + 62 = 7135°$, at which temperature the 7101 cubic feet of mixed gases would become $7101 \times \frac{458 + 7135}{458 + 62} = 103675$ cubic feet if expansion were permitted, but being confined by a vessel of 8115 cubic feet capacity the pressure by the law of Marriotte would be $103675 \div 8115 = 12 \cdot 65$ atmospheres, instead of $22 \cdot 2$, as with pure oxygen.

Again, 1 lb. of hydrogen requires by (57) 8 lbs. of oxygen for its combustion, and the capacity of the closed vessel would be $189 \cdot 7 + (11 \cdot 88 \times 8) = 284 \cdot 74$ cubic feet. After combustion there would be 9 lbs. of vapour of water, whose volume would be $21 \cdot 07 \times 9 = 189 \cdot 7$ cubic feet. The combustion of 1 lb. of hydrogen (57) develops 62,535 units of heat, and the specific heat of vapour of water with constant volume being $\cdot 364$ by Table 5, that amount of heat would raise the temperature of 9 lbs. of vapour $62535 \div (\cdot 364 \times 9) = 19088°$, or to $19088 + 62 = 19150°$. At that temperature the $189 \cdot 7$ cubic feet of vapour would become $189 \cdot 7 \times \frac{458 + 19150}{458 + 62} = 7151$ cubic feet if expansion were permitted; hence in a closed vessel of $284 \cdot 74$ cubic feet capacity the pressure would become $7151 \div 284 \cdot 74 = 25 \cdot 1$ atmospheres.

(418.) "*Evaporation at Low Pressures of Air.*"—The rule and experiments in (186) give the evaporation at natural temperatures, with calm air at the normal pressure of say 30 inches of mercury in the barometer, and it will be important to note the effect of reduced pressure on the rate of evaporation. The experiments of Daniell show that the evaporation of water is nearly inversely proportioned to the pressure, so that at half the normal pressure the evaporation would be doubled, &c. With a vacuum as nearly perfect as could be obtained, the pressure being $\frac{1}{434}$th of the normal pressure, the evaporation was about seventy times that due to 30 inches.

With pressures in the ratio

1	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{1}{32}$	$\frac{1}{64}$

the rate of evaporation, if it were simply in inverse ratio, would of course be

1	2	4	8	16	32	64

but experiment gave

1	2	4·4	7·1	12	20	32

We found in (190) that with a gale the ratio for water was 12·4 times the evaporation with calm air. The above experiments show that the same rate of evaporation might be obtained with calm air by reducing the pressure to $\frac{1}{16}$th of the normal pressure, or say $30 \div 16 = 1 \cdot 9$ inch of mercury.

(419.) "*Absorption of Gases by Liquids.*"—When gases such as carbonic acid or atmospheric air are in contact with a surface of water or other liquid, a certain *volume* of gas is absorbed, dependent on the nature of the liquid and of the gas, and on the temperature.

In general the amount absorbed decreases with increase of temperature, thus if water or ale at 32°, saturated with carbonic acid gas, be warmed to 68°, it could hold only half the amount of gas, and the rest is set free as bubbles, and collects as froth at the surface. Similar results are given by other gases: thus at 32° a cubic foot of water absorbs of

Nitrogen.	Oxygen.	Carbonic Acid.	Carbonic Oxide	Sulphuretted Hydrogen.	Sulphurous Acid.	Atmospheric Air.
·024	·0411	1·797	·0329	4·371	68·86	·0247

cubic foot; but at 68° these volumes are reduced to

·014	·0286	·901	·0231	2·905	36·22	·0170

cubic feet respectively. It is remarkable that the amount absorbed is a question of *volume* of gas and not of *weight*, that is to say, the volume absorbed is the same whatever the pressure or density of the gas. Aerated waters are made on this principle; the liquid is saturated with carbonic acid gas under a high pressure, which of course escapes with effervescence when the pressure is relieved.

(420.) "*Ice-houses.*"—For the preservation of ice collected during the winter season ice-houses are usually made in the form of a large conical pit from 12 to 15 feet diameter at the top, and 20 feet deep, covered with a roof thickly thatched with straw, and lined throughout with a thick layer of the same material.

Fig. 1.

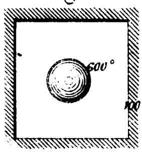


Fig. 2.

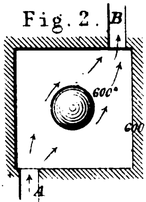


Fig. 3.

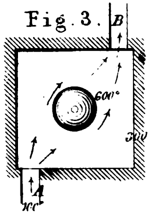


Fig. 4.

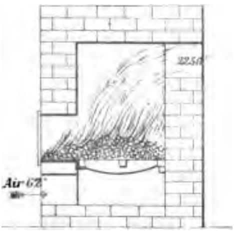


Fig. 5.

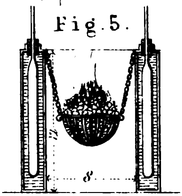


Fig. 6.

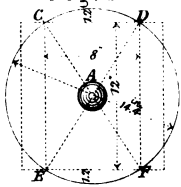


Fig. 7.

Fig. 9.

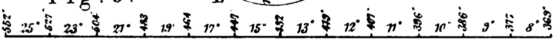


Fig. 10.

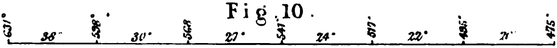
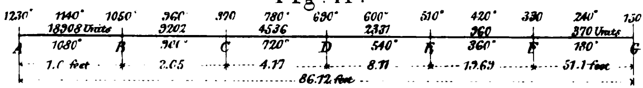
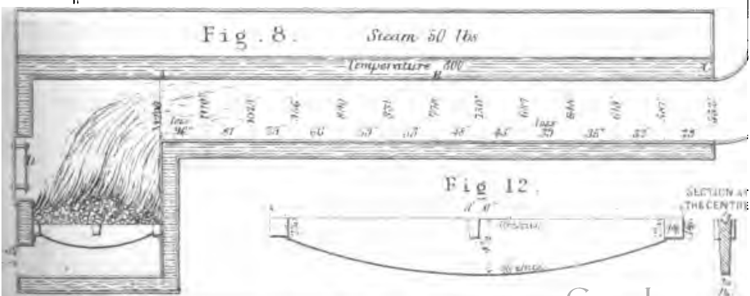


Fig. 11.





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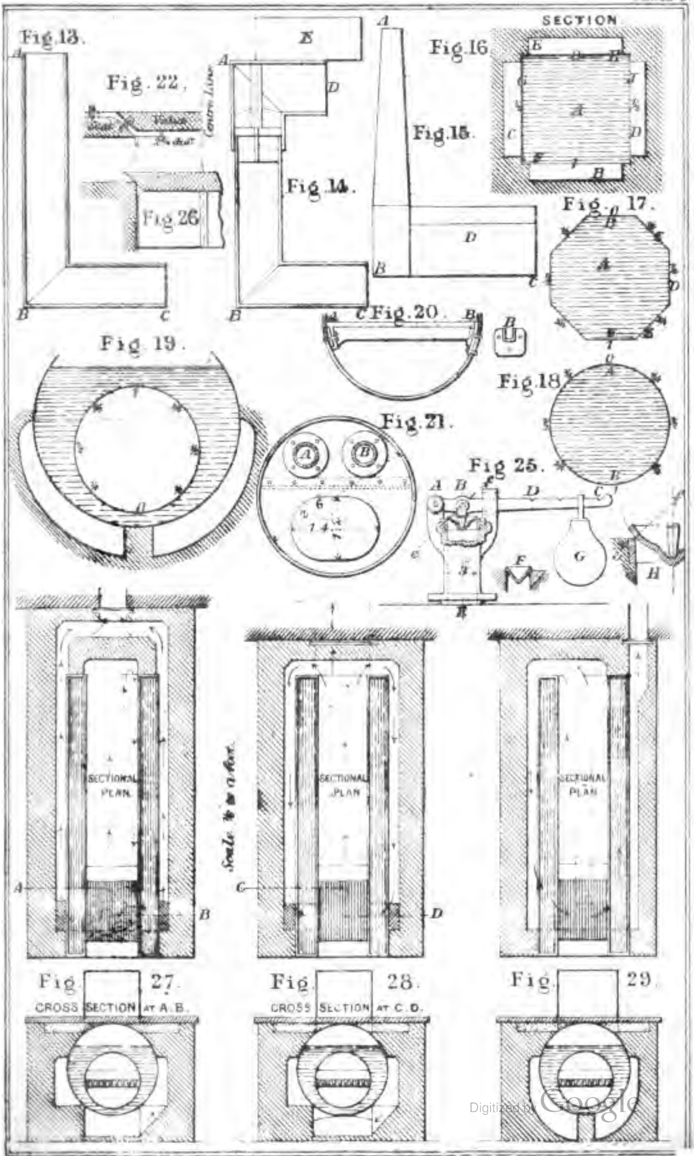

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Fig. 34.

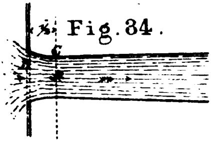


Fig. 31.

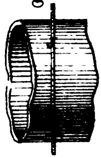


Fig. 30.

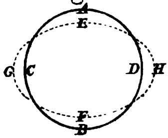


Fig. 32.

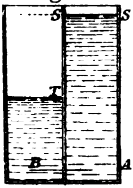


Fig. 33.

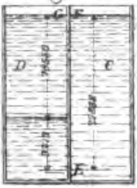
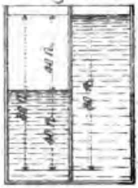
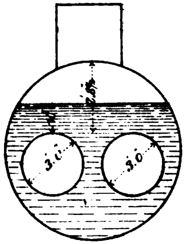


Fig. 35.





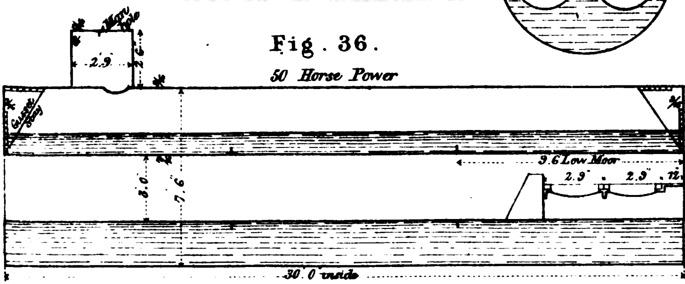
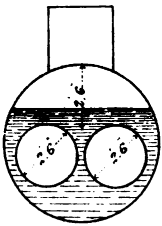


Fig. 36.

50 Horse Power



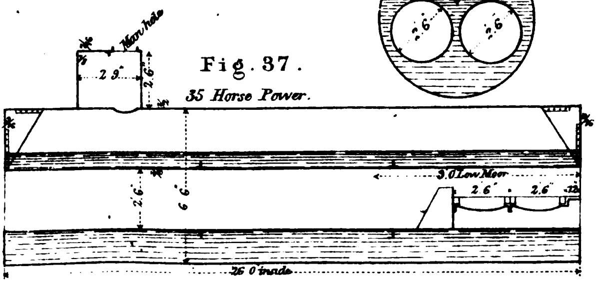


Fig. 37.

35 Horse Power.

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For 35 and 50 H.P. See Plate 3.

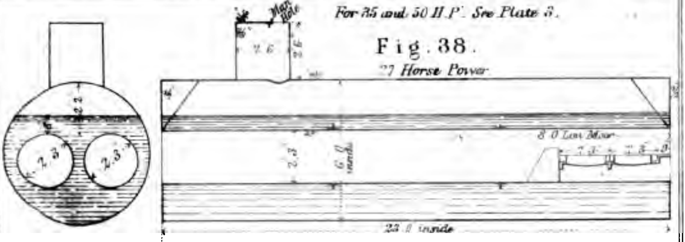

Fig. 38.

27 Horse Power

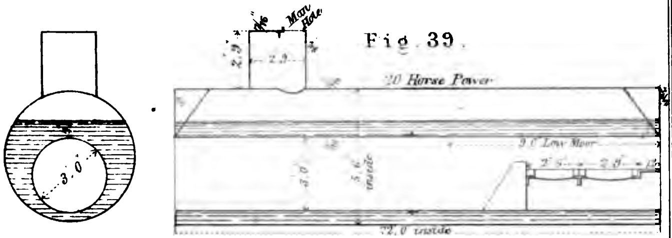

Fig. 39.

20 Horse Power

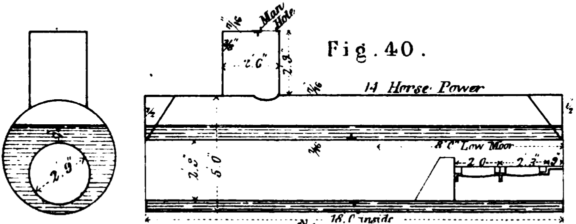

Fig. 40.

14 Horse Power

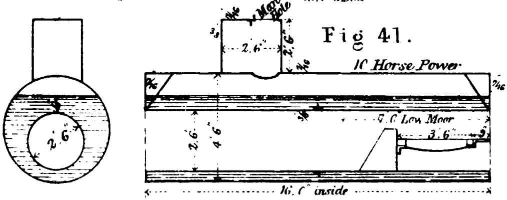

Fig. 41.

10 Horse Power

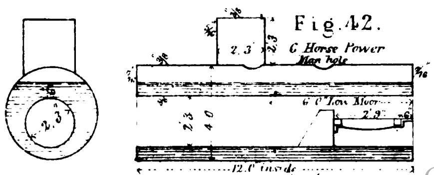

Fig. 42.

6 Horse Power

For 4 H.P. See Plate 5.

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Fig. 43.

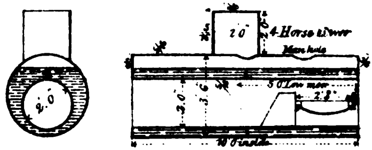


Fig. 44.

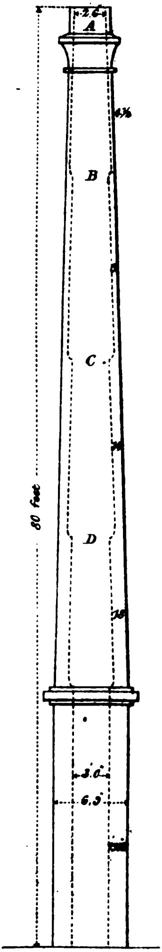


Fig. 47.

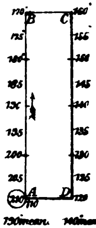
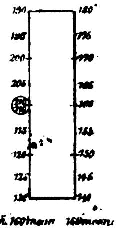


Fig. 48.



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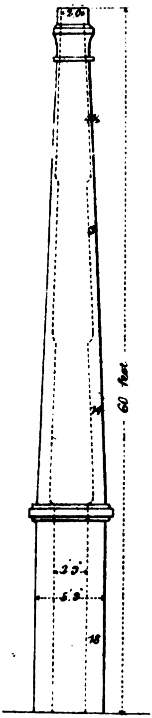

Fig. 46.

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PLATE 6.

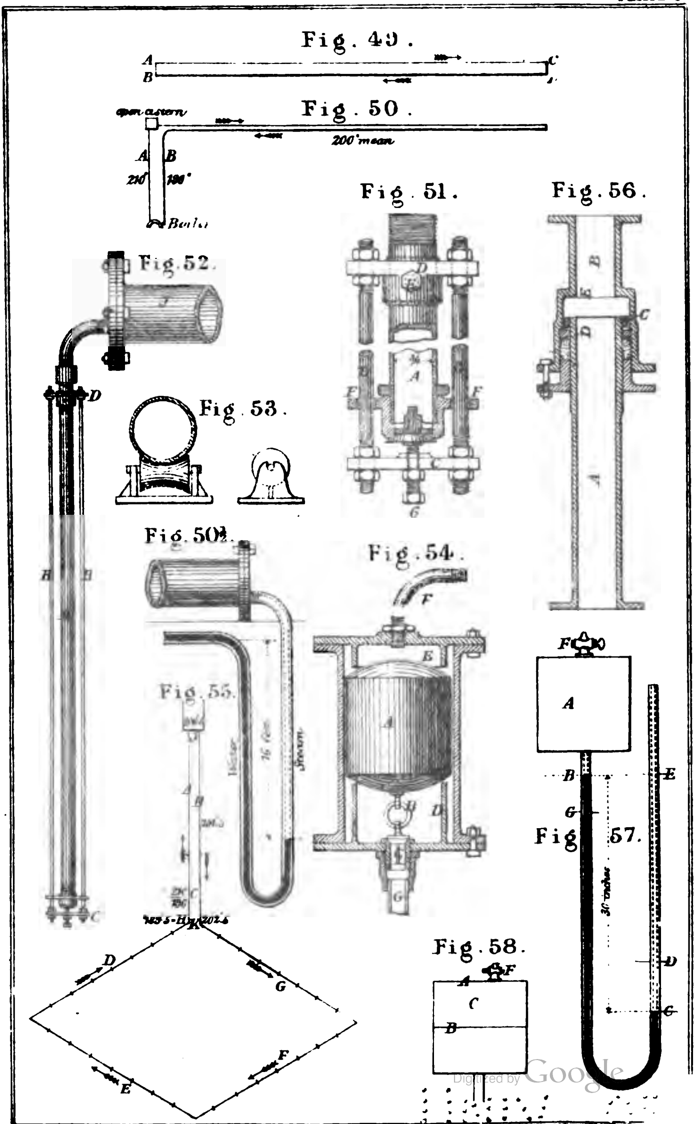

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Fig. 59.

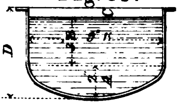

Fig. 60.

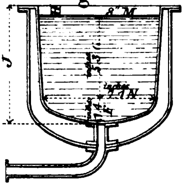

Fig. 61.

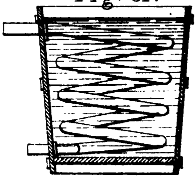

Fig. 62.

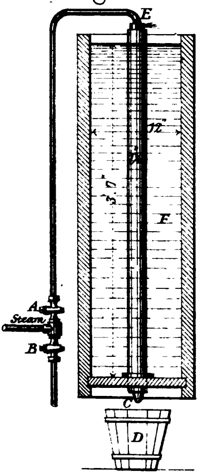

Fig. 63.

SECTION.

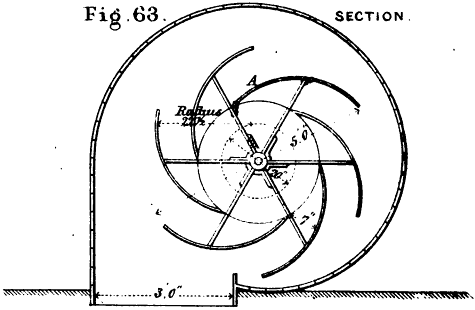

Fig. 64.

SECTIONAL PLAN.

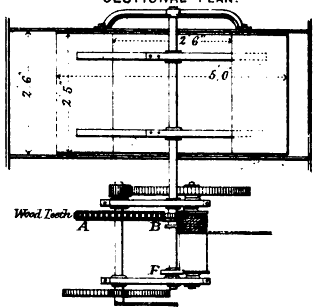

Fig. 67.

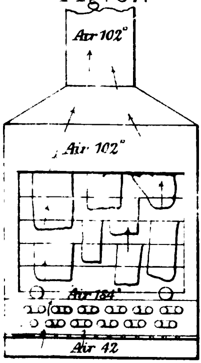

Fig. 65.

DIAGRAM OF GEARING.

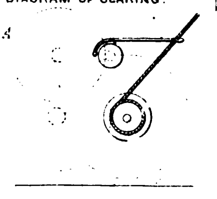

Fig. 66.

SIDE ELEVATION OF FRAMING

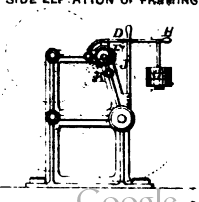

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Fig. 68.
LONGITUDINAL SECTION.

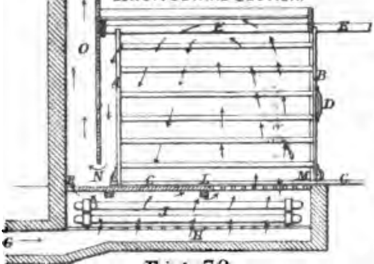


Fig. 69.
CROSS SECTION.

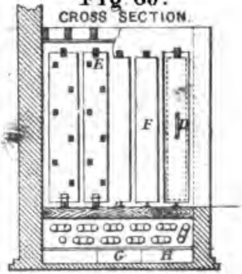
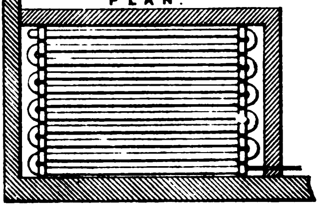


Fig. 70.
PLAN.



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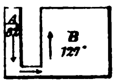

Fig. 72.

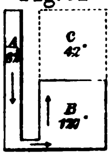

Fig. 73.

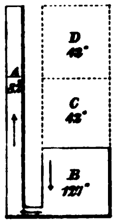

Fig. 74.
Loss by Radiation.

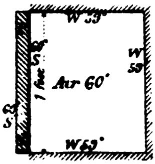

Fig. 75.
Loss by Cold Air

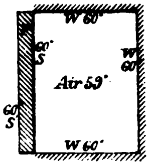

Fig. 76.
Loss by Conduction

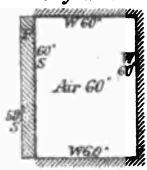

Fig. 77.
Loss by R, A, C

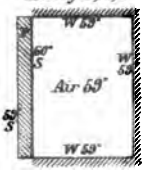

Fig. 78.

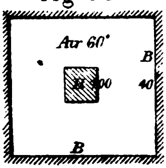

Fig. 79.

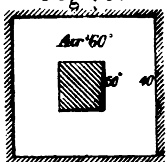

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Fig. 80.

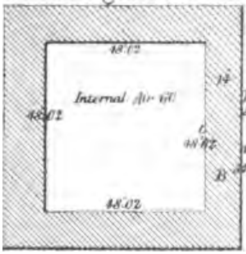
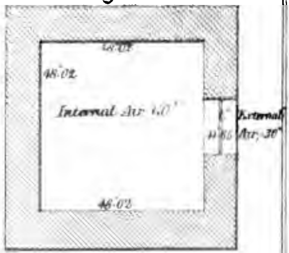


Fig. 81

Fig. 82.



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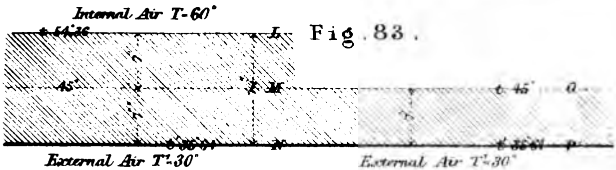

Fig. 84.

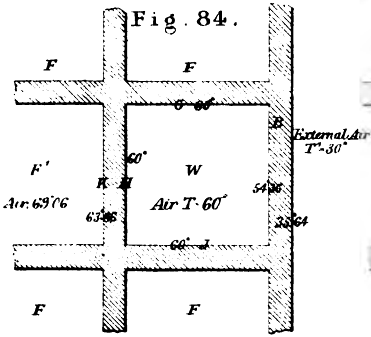

Fig. 85.

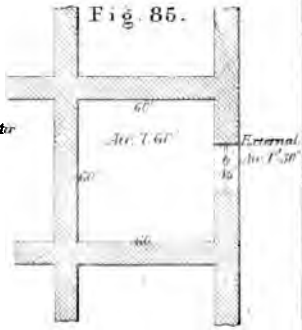

Fig. 85½

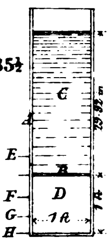

Fig. 86.

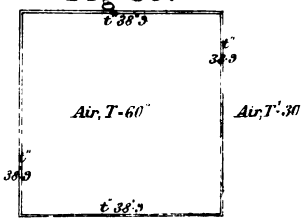

Fig. 86½

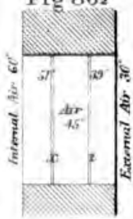

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Fig. 87.

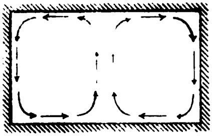


Fig. 89.

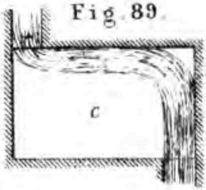


Fig. 88.

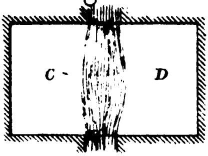


Fig. 90.

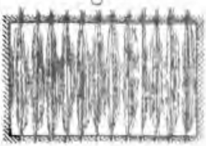
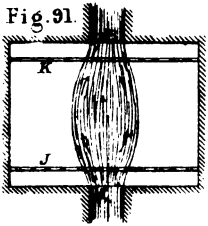


Fig. 91.



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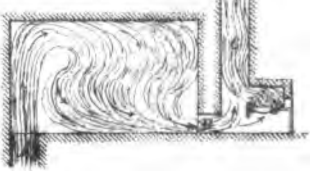

Fig. 93.

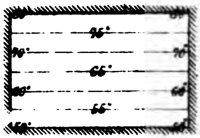

Fig. 94.

Fig. 95.

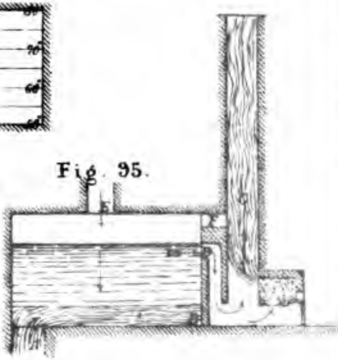

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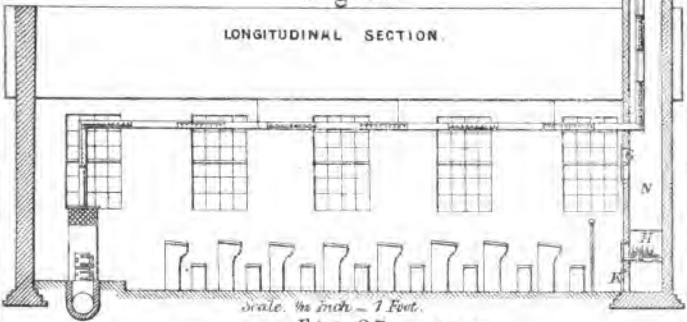
Fig. 96.

LONGITUDINAL SECTION.



Scale. 1/16 Inch = 1 Foot.

Fig. 97. PLAN.

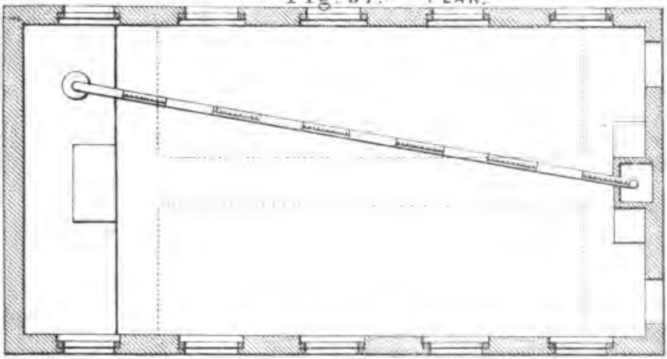
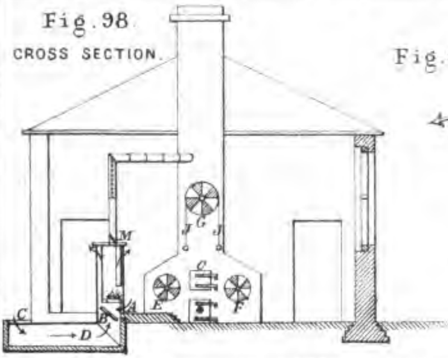


Fig. 98

CROSS SECTION.



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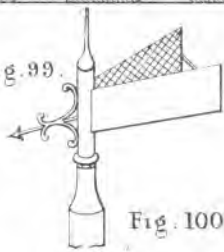

Fig. 100.

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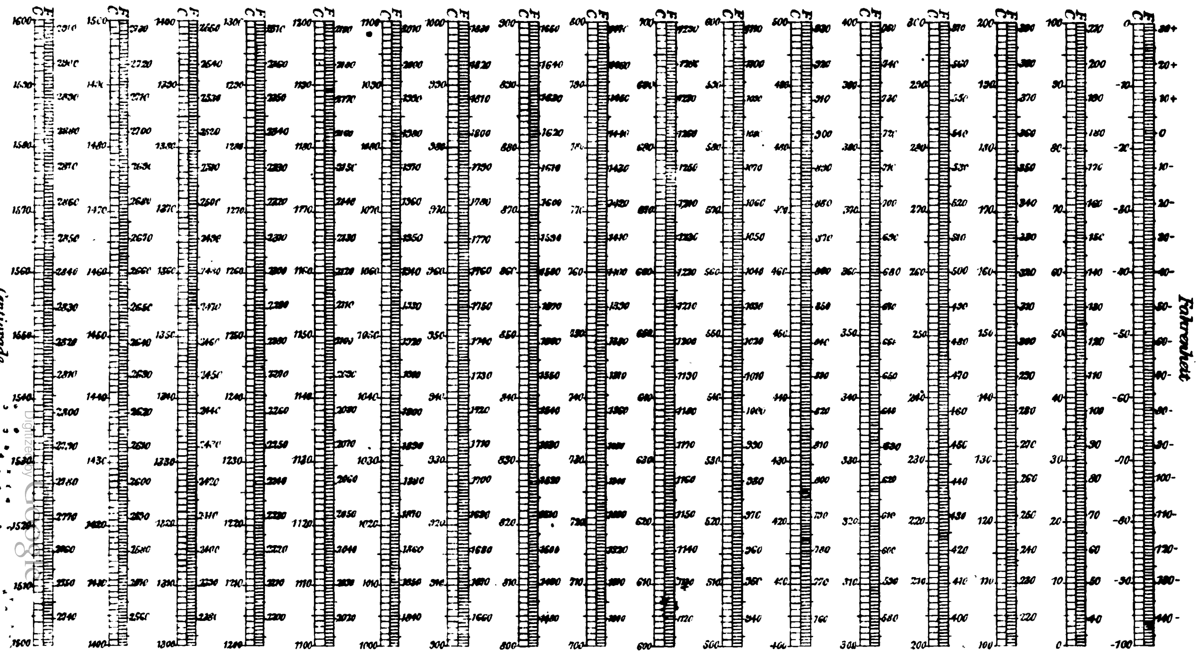
TABLE OF THE CORRESPONDENCE OF THE FAHRENHEIT AND CENTIGRADE THERMOMETERS.



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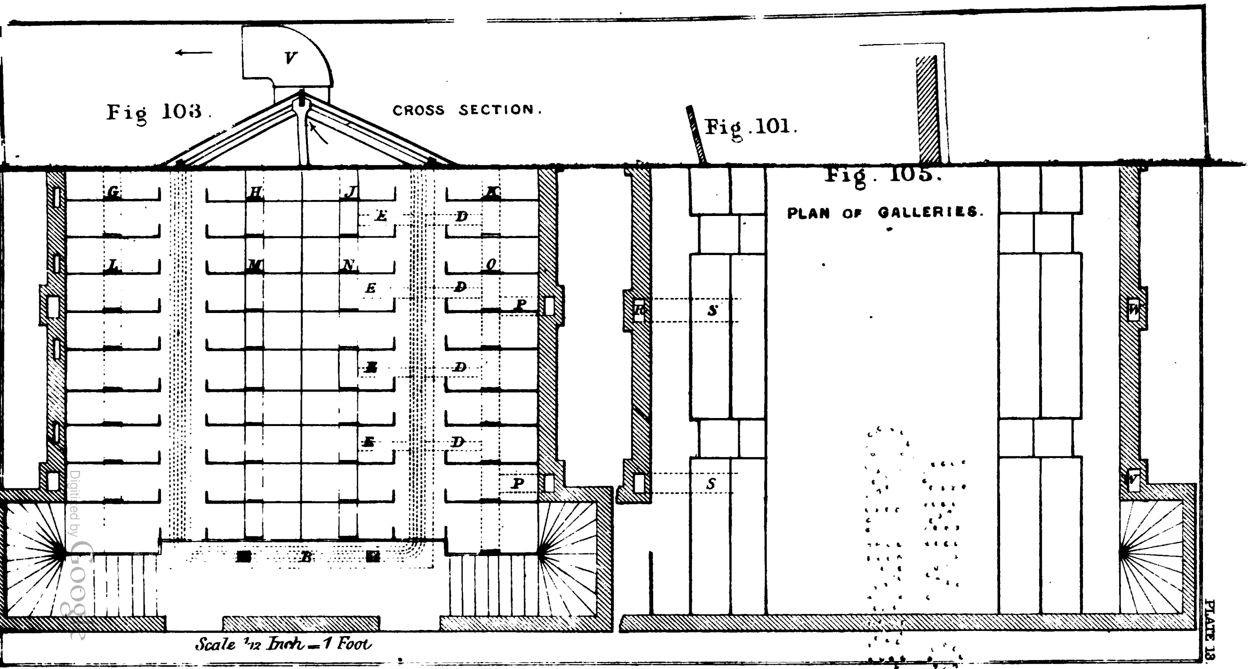
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CROSS SECTION.
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D
P
B
Fig. 101.
Fig. 105.
PLAN OF GALLERIES.
S
W
Scale 1/12 Inch = 1 Foot

ig. 110.
DINAL SECTION.

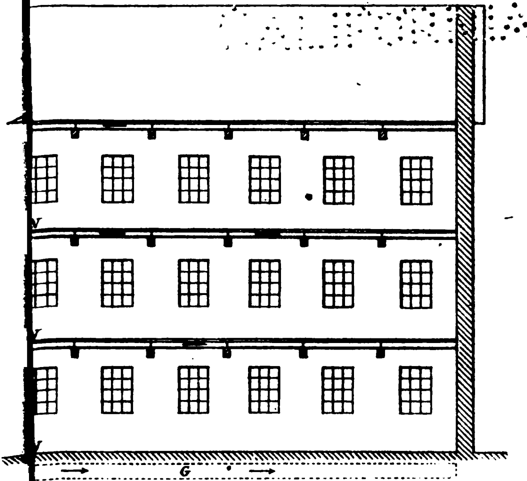
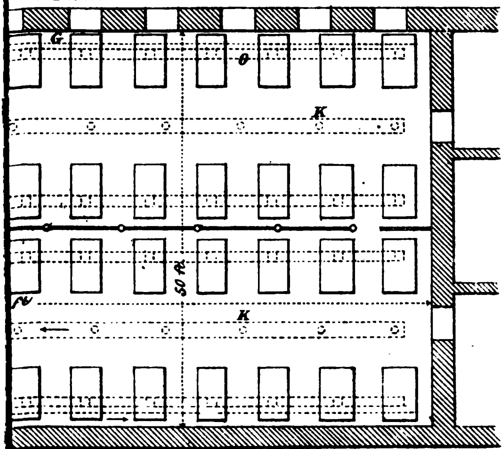
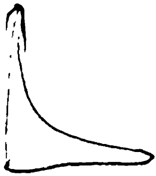


Fig. 109.
PLAN.





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